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# TABLE OF CONTENTS

|                                                                                                                                                    |          |
|----------------------------------------------------------------------------------------------------------------------------------------------------|----------|
| <b>FOREWORD.....</b>                                                                                                                               | <b>1</b> |
| <b>SECTION I: STRUCTURAL ANALYSIS</b>                                                                                                              |          |
| EXPERIMENTING FOR KNOWING: EVOLUTION OF A CONCEPT IN TECHNICAL SCIENCE.....                                                                        | 5        |
| <i>Alessandro FREDDI</i>                                                                                                                           |          |
| EXPERIMENTAL AND NUMERICAL STUDY ON THE BENDING STRENGTH OF A T-CORE SANDWICH PANEL.....                                                           | 9        |
| <i>Adriana SANDU, Ștefan SOROHAN, Marin SANDU, Dan Mihai CONSTANTINESCU, Emil NUȚU</i>                                                             |          |
| IMPACT ON A MICRO-REINFORCED UHPC: EXPERIMENTAL STUDIES VS. NUMERICAL MODELING.....                                                                | 11       |
| <i>Astrid KOWITZ, Cheng-Chieh WU, Falk HILLE, Rosemarie HELMERICH, Daniel KADOKE, Klaus-Peter GRÜNDER, Stephan HAUSER, Henrik SCHWARZINGER</i>     |          |
| A METHOD FOR STRAIN ANALYSES OF SURFACES WITH CURVED BOUNDARIES BASED ON MEASURED DISPLACEMENT FIELDS .....                                        | 13       |
| <i>Thomas LEHMANN, Martin STOCKMANN, Jörn IHLEMANN</i>                                                                                             |          |
| COMPARISON OF FRACTURE PROPERTIES OF STEEL PIPES .....                                                                                             | 15       |
| <i>Lubomír GAJDOŠ, Martin ŠPERL</i>                                                                                                                |          |
| EXPERIMENTAL INVESTIGATION OF RESTITUTION COEFFICIENT FOR HYBRID COMPOSITES.....                                                                   | 17       |
| <i>Marcin OBSZAŃSKI, Marek MATYJEWSKI</i>                                                                                                          |          |
| DAMAGE MONITORING OF DIFFERENT CONCENTRATION CARBON NANOTUBE/EPOXY GLASS FIBER REINFORCED COMPOSITES UNDER QUASI-STATIC INCREMENTAL LOADINGS ..... | 19       |
| <i>Ilona LAZARIDOU, Stavros K. KOURKOULIS, Nikolaos D. ALEXOPOULOS</i>                                                                             |          |
| DESIGN OF COMMERCIAL HEXAGONAL HONEYCOMBS OF EQUAL OUT-OF-PLANE SHEAR MODULI..                                                                     | 21       |
| <i>Ștefan SOROHAN, Dan Mihai CONSTANTINESCU, Marin SANDU, Adriana SANDU</i>                                                                        |          |
| A SMALL-SCALE TEST BRIDGE FOR MEASUREMENT- AND MODEL-BASED STRUCTURAL ANALYSIS ..                                                                  | 23       |
| <i>Cheng-Chieh WU, Daniel KADOKE, Michael FISCHER, Harald KOHLHOFF, Sven WEISBRICH, Frank NEITZEL</i>                                              |          |
| FATIGUE DAMAGE PREDICTION OF NOTCHED SAMPLES.....                                                                                                  | 25       |
| <i>Liviu Daniel PÎRVULESCU, Anghel CERNESCU</i>                                                                                                    |          |
| STIFFNESS DETERMINATION IN STEEL SPECIMENS ORIGINATING FROM NEW AND DISCARDED TRAMWAY RAILS BY MEANS OF NANOINDENTATION.....                       | 27       |
| <i>Valentin JAGSCH, Patricia HASSLINGER, Olaf LAHAYNE, Christian HELLMICH, Stefan SCHEINER</i>                                                     |          |
| NUMERICAL AND EXPERIMENTAL INVESTIGATION OF CONTACT LENGTH DURING ORTHOGONAL CUTTING .....                                                         | 29       |
| <i>Szabolcs BEREZVAI, Tamas G. MOLNAR, Attila KOSSA, Daniel BACHRATHY, Gabor STEPAN</i>                                                            |          |
| ROOT ITERATION METHOD FOR MODAL PARAMETER IDENTIFICATION.....                                                                                      | 31       |
| <i>Zoltan LOVASZ, Zoltan DOMBOVARI</i>                                                                                                             |          |
| APPRECIABLE CROSS SECTION REDUCTION UPON MODERATE INTRADOS BENDING STRESS INCREASE IN CURVED BEAMS.....                                            | 33       |
| <i>Antonio STROZZI, Enrico BERTOCCHI, Sara MANTOVANI</i>                                                                                           |          |
| EXPERIMENTAL APPROACH TO THE VERIFICATION OF NUMERICAL MODEL OF HYBRID MACHINE FOR CONSTRUCTION WORKS.....                                         | 35       |
| <i>Jakub ANDRUSZKO, Damian DERLUKIEWICZ, Mariusz STAŃCO</i>                                                                                        |          |
| NUMERICAL AND EXPERIMENTAL ANALYSIS OF THE RUBBER BUMPER STIFFNESS.....                                                                            | 37       |
| <i>Mariusz STAŃCO, Paulina DZIAŁAK</i>                                                                                                             |          |

|                                                                                                                                                                 |    |
|-----------------------------------------------------------------------------------------------------------------------------------------------------------------|----|
| IMPACT RESPONSE OF SANDWICH PANELS WITH POLYURETHANE AND POLYSTYRENE CORE AND COMPOSITE FACESHEETS .....                                                        | 39 |
| <i>Oana Alexandra MOCIAN, Dan Mihai CONSTANTINESCU, Marin SANDU, Ștefan SOROHAN, Dorin ROȘU, Michael FEUCHTER</i>                                               |    |
| MEASURING THE AXIAL FORCE ON TURBO-JET ENGINE ROTOR .....                                                                                                       | 41 |
| <i>Nenad KOLAREVIĆ, Miloš STANKOVIĆ, Stevan CRNOJEVIĆ, Dušan MICKOVIĆ, Marko MILOŠ</i>                                                                          |    |
| METHODOLOGY OPTIMIZATION TESTS OF COMPOSITE ROTOR BLADES OF HELICOPTERS IN EXTREME CONDITIONS .....                                                             | 43 |
| <i>Časlav B. MITROVIĆ, Nebojša PETROVIĆ, Goran VOROTOVIĆ, Mirjana OPAČIĆ</i>                                                                                    |    |
| EXPERIMENTAL STRESS ANALYSIS OF A CANTILEVER BEAM .....                                                                                                         | 45 |
| <i>Ionuț Daniel GEONEA</i>                                                                                                                                      |    |
| <b>SECTION II: MATERIALS CHARACTERIZATION AND TESTING</b>                                                                                                       |    |
| EXPERIMENTAL DYNAMICAL ANALYSIS OF SPECIMENS' MATERIAL PROPERTIES MANUFACTURED BY ADDITIVE TECHNOLOGIES .....                                                   | 49 |
| <i>Bálint Ádám KOVÁCS, Peter FICZERE, Lajos BORBÁS</i>                                                                                                          |    |
| INVESTIGATION OF THE SHRINKAGE BEHAVIOR OF CONCRETE CONSIDERING THE MOISTURE CONTENT .....                                                                      | 51 |
| <i>Stefan SMANIOTTO, Martin DREXEL, Günter HOFSTETTER</i>                                                                                                       |    |
| DAMAGE OBSERVATION IN GLASS FIBER REINFORCED COMPOSITES VIA M-TOMOGRAPHY .....                                                                                  | 53 |
| <i>Zvonimir TOMIČEVIĆ, Amine BOUTERF, Robert SURMA, François HILD</i>                                                                                           |    |
| CHARACTERIZATION OF A CORK-RUBBER COMPOSITE USING ADVANCED MATERIAL MODELS.....                                                                                 | 55 |
| <i>Jiří KOCÁB, Radek KOTTNER, Attila KOSSA</i>                                                                                                                  |    |
| YIELDING AND STRAIN LOCALIZATION EFFECTS IN GUM METAL - A UNIQUE TI ALLOY - INVESTIGATED BY DIGITAL IMAGE CORRELATION AND INFRARED THERMOGRAPHY .....           | 57 |
| <i>Elżbieta A. PIECZYSKA, Karol GOLASIŃSKI, Michał MAJ, Maria STASZCZAK, Zbigniew L. KOWALEWSKI, Tadahiko FURUTA, Shigeru KURAMOTO</i>                          |    |
| TOUGHNESS TESTS OF TOOL STEELS FOR WORK AT LOW TEMPERATURE .....                                                                                                | 59 |
| <i>Željko DOMAZET, Francisko LUKŠA, Petra BAGAVAC</i>                                                                                                           |    |
| MICRO- AND MACROSCOPIC CHARACTERIZATION OF GRANITE .....                                                                                                        | 61 |
| <i>Josef EBERHARDSTEINER, Sophie SCHMID, Olaf LAHAYNE, Roland REIHSNER, Bernhard PICHLER</i>                                                                    |    |
| INFRARED THERMOGRAPHY OF STEEL STRUCTURE BY FFT .....                                                                                                           | 63 |
| <i>Petra BAGAVAC, Lovre KRSTULOVIĆ-OPARA, Željko DOMAZET</i>                                                                                                    |    |
| MECHANICAL PROPERTIES OF COMPOSITE STRUCTURAL PARTS FOR MACHINE TOOLS .....                                                                                     | 65 |
| <i>Milan RŮŽIČKA, Viktor KULÍŠEK, Pavel VRBA, Jan SMOLÍK, Zdeněk POŠVÁŘ</i>                                                                                     |    |
| DETECTION OF SURFACE DAMAGE USING RESISTANCE TOMOGRAPHY IN THIN GRAPHENE LAYER .                                                                                | 67 |
| <i>Marek STEPNOWSKI, Paweł PYRZANOWSKI</i>                                                                                                                      |    |
| WATER PERMEABILITY OF HOT MIX ASPHALT .....                                                                                                                     | 69 |
| <i>Daniel STEINER, Bernhard HOFKO</i>                                                                                                                           |    |
| ON THE COMPRESSIBILITY OF RUBBER: EXPERIMENTS AND THEORETICAL CONSIDERATIONS .....                                                                              | 71 |
| <i>Robert PLACHY, Stefan SCHEINER, Krzysztof W. LUCZYNSKI, Armin HOLZNER, Christian HELLMICH</i>                                                                |    |
| EFFECT OF AS-CAST SURFACE LAYER ON ALUMINIUM ALLOY FATIGUE STRENGTH .....                                                                                       | 73 |
| <i>Sebastian POMBERGER, Martin LEITNER, Michael STOSCHKA</i>                                                                                                    |    |
| EARLY-AGE CHARACTERIZATION AND MULTI-SCALE MODELING OF POLYMER-MODIFIED CEMENT PASTES.....                                                                      | 75 |
| <i>Luise GÖBEL, Markus KÖNIGSBERGER, Andrea OSBURG, Bernhard PICHLER</i>                                                                                        |    |
| MICROMECHANICAL ANALYSIS OF FIBER SPATIAL DISTRIBUTION INFLUENCE IN UNIDIRECTIONAL COMPOSITE CROSS-SECTION ON OVERALL RESPONSE IN TERMS OF TENSILE CURVES ..... | 77 |
| <i>Tomáš KROUPA, Hana ZEMČÍK, Robert ZEMČÍK</i>                                                                                                                 |    |

|                                                                                                                                                                                                                                                                          |     |
|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----|
| VALIDATION OF IDENTIFIED MATERIAL PARAMETERS OF RUBBER USING AN ARCAN SHEAR TEST .79<br><i>Radek KOTTNER, Jan HECZKO, Jan KRYSTEK</i>                                                                                                                                    |     |
| A NEW SURFACE PREPARATION METHOD FOR MICROSCOPIC DIGITAL IMAGE CORRELATION APPLICATIONS.....81<br><i>Edgar PERETZKI, Martin STOCKMANN, Thomas LEHMANN, Jörn IHLEMANN</i>                                                                                                 | 81  |
| DYNAMIC TESTING OF A SPECIMEN SETUP FOR COMBINED HIGH PRECISION UNIAXIAL TENSION-COMPRESSION TESTS OF RUBBER .....83<br><i>Lars KANZENBACH, Erik OELSCH, Thomas LEHMANN, Jörn IHLEMANN</i>                                                                               | 83  |
| ELASTIC-PLASTIC CHARACTERIZATION OF 2024-T351 ALUMINUM MATERIAL BASED ON RING COMPRESSION TEST, BENDING TEST AND UNIAXIAL TENSION/COMPRESSION TESTS.....85<br><i>Attila KOSSA, Gabor STEPAN</i>                                                                          | 85  |
| INVESTIGATING THE EFFECTS OF SEVERE SHOT PEENING ON PURE IRON .....87<br><i>Sara BAGHERIFARD, Mauro Filippo MOLLA, Mario GUAGLIANO</i>                                                                                                                                   | 87  |
| S-N CURVE OF COMPOSITE SPECIMENS DETERMINED IN TENSION-TENSION FATIGUE TESTING.....89<br><i>Ionel IACOB, Ionel CHIRICĂ, Elena-Felicia BEZNEA</i>                                                                                                                         | 89  |
| COMPARISONS BETWEEN SOME COMPOSITE MATERIALS REINFORCED WITH HEMP FIBERS .....91<br><i>Cosmin Mihai MIRIȚOIU, Cristian Oliviu BURADA, Marius Marinel STĂNESCU, Dumitru BOLCU, Adriana PĂDEANU, Alexandru BOLCU</i>                                                       | 91  |
| STATIC AND DYNAMIC MECHANICAL PROPERTIES OF A NEW GREEN RESIN BASED ON DAMMAR ....93<br><i>Cosmin Mihai MIRIȚOIU, Marius Marinel STĂNESCU, Alexandru BOLCU, Dumitru BOLCU, Cristian Oliviu BURADA, Valeriu IONICĂ</i>                                                    | 93  |
| EXPERIMENTAL STUDIES ON UNIAXIAL AND ECHIBIAXIAL TENSILE TESTS APPLIED TO PLASTIC MATERIALS SHEETS .....95<br><i>Adrian PASCU, Mihaela OLEKSIK, Nicolae ROȘCA, Eugen AVRIGEAN, Valentin OLEKSIK</i>                                                                      | 95  |
| STATIC AND DYNAMIC TESTS FOR DETERMINING THE CHARACTERISTIC PERFORMANCE OF MECHANICAL SPLICES IN REINFORCED CONCRETE .....97<br><i>Ovidiu VASILE, Oana TONCIU</i>                                                                                                        | 97  |
| EXPERIMENTAL DETERMINATION OF FATIGUE PROPERTIES AND E-N CURVES FOR S355J2+N STEEL GRADE.....99<br><i>Vladimir MILOVANOVIĆ, Miroslav ŽIVKOVIĆ, Gordana JOVIČIĆ</i>                                                                                                       | 99  |
| EXPERIMENTAL DETERMINATION OF STRAIN HARDENING EXPONENT FOR STREX 700..... 101<br><i>Aleksandar DIŠIĆ, Miroslav ŽIVKOVIĆ, Vladimir MILOVANOVIĆ</i>                                                                                                                       | 101 |
| DETERMINING THE EMISSIVITY FACTOR OF THERMOPLASTIC MATERIALS FOR A THERMOPLASTIC BENDING PROCESS ..... 103<br><i>Martin KLINGENBRUNNER, Julia PLATZER, Friedrich BLEICHER</i>                                                                                            | 103 |
| THREE-POINT BENDING BEHAVIOUR OF ORGANIC MODIFIED COMPOSITES..... 105<br><i>Cristian MUNTENIȚĂ, Marian Sebastian DRĂGHICI, Violeta SAVA, Vasile BRIA, Adrian CÎRCIUMARU, Iulia GRAUR</i>                                                                                 | 105 |
| <b>SECTION III: BIOMECHANICS</b>                                                                                                                                                                                                                                         |     |
| EVALUATION OF HUMAN KNEE FLEXION-EXTENSION ON INCLINED TREADMILL USING WEARABLE SENSORS ..... 109<br><i>Alin PETCU, Daniela TARNIȚĂ, Ionuț Daniel GEONEA, Raluca MALCIU</i>                                                                                              | 109 |
| EXPLORING EXPERIMENTALLY THE POTENTIALITIES OF PLATELET RICH PLASMA IN TREATING PELVIC ORGAN PROLAPSE ..... 111<br><i>Stavros K. KOURKOULIS, Eleftheria L. CHRYSANTHOPOULOU, Vasileios PERGIALIOTIS, Despina N. PERREA, Stergios K. DOUMOUCHTSIS, Chara SPILIOPOULOU</i> | 111 |
| BONE STRUCTURE INVESTIGATION IN A CASE OF AUTOSOMAL RECESSIVE OSTEOPETROSIS..... 113<br><i>Francesca COSMI, Natalia MAXIMOVA</i>                                                                                                                                         | 113 |

|                                                                                                                                                     |     |
|-----------------------------------------------------------------------------------------------------------------------------------------------------|-----|
| DISPLACEMENT-ENCODED MRI REVEALS ALTERED STRAIN PATTERNS IN ENZYMATICALLY DEGRADED CARTILAGE DURING CYCLIC COMPRESSION .....                        | 115 |
| <i>Maria-Ioana PASTRAMĂ, Lianne ZEVENBERGEN, Nele FAMAIEY, Willy GSELL, Corey P. NEU, Uwe HIMMELREICH, Ilse JONKERS</i>                             |     |
| NONLINEAR DYNAMICS OF HUMAN KNEE JOINT DURING JUMPS .....                                                                                           | 117 |
| <i>Daniela TARNIȚĂ, Dan MARGHITU, Alin PETCU</i>                                                                                                    |     |
| PERFORMANCE MEASUREMENT OF SAILORS WITH CUSTOM HIKING BENCH AND MOTION TRACKING SYSTEM.....                                                         | 119 |
| <i>Bernadett KISS, Rita KISS</i>                                                                                                                    |     |
| ANALYSING HUMAN BALANCE RECOVERY ACTION USING CALCULATED TORQUES OF A DOUBLE PENDULUM MODEL .....                                                   | 121 |
| <i>Bernadett KISS, Balint PETRO, Rita KISS</i>                                                                                                      |     |
| INVESTIGATION OF THE DESIGN ASPECTS OF RUNNING SUITS APPLYING MOTION ANALYSIS .....                                                                 | 123 |
| <i>Zsófia PÁLYA, Anna GULYÁS, Marianna HALÁSZ, Rita KISS</i>                                                                                        |     |
| HUMAN BALANCING IN THE SAGITTAL PLANE .....                                                                                                         | 125 |
| <i>Csenge A. MOLNÁR, Ambrus ZELEI, Tamás INSPERGER</i>                                                                                              |     |
| 3D-PRINTED ANKLE-FOOT ORTHOSIS: A DESIGN METHOD .....                                                                                               | 127 |
| <i>Alberto DAL MASO, Francesca COSMI</i>                                                                                                            |     |
| AUTOMATIC GUIDANCE OF BIOPSY CATHETER TO PERIPHERAL AIRWAYS TARGETS USING A NOVEL ROBOTIC SYSTEM.....                                               | 129 |
| <i>Lucian Gheorghe GRUIONU, Cătălin CONSTANTINESCU, Andreea ȘOIMU-IACOB, Cătălin CIOBÎRCĂ, Anca UDRIȘTOIU, Ștefan Dan PASTRAMĂ, Gabriel GRUIONU</i> |     |
| <b>SECTION IV: PRACTICAL APPLICATIONS/CASE STUDIES/ INSTRUMENTATION</b>                                                                             |     |
| OPTIMIZATION OF LASER BEAM CUTTING PARAMETERS OF HIGH DENSITY COMPOSITE FIBRE CEMENT FLAT BOARD .....                                               | 133 |
| <i>László MOLNÁR, Tamás CSISZÉR, Lajos BORBÁS, Levente László BOGNÁR</i>                                                                            |     |
| DYNAMIC TEST OF A BRIDGE OVER THE DANUBE-BLACK SEA CHANNEL AT AGIGEA.....                                                                           | 135 |
| <i>Cristian Lucian GHINDEA, Radu Iuliu CRUCIAT, Ionuț Radu RĂCĂNEL</i>                                                                              |     |
| SUPERSONIC TEST CASES AT HIGH ANGLES OF ATTACK .....                                                                                                | 137 |
| <i>Dijana DAMLJANOVIĆ, Đorđe VUKOVIĆ, Jovan ISAKOVIĆ, Marko MILOŠ</i>                                                                               |     |
| CONCEPT OF A GAS-SENSITIVE NANO AERIAL ROBOT SWARM FOR INDOOR AIR QUALITY MONITORING .....                                                          | 139 |
| <i>Patrick P. NEUMANN, Dino HÜLLMANN, Matthias BARTHOLMAI</i>                                                                                       |     |
| PASSIVE RFID TRANSPONDER – LOW-POWER MULTI-SENSOR INTERFACE FOR STRUCTURAL HEALTH MONITORING IN CONCRETE .....                                      | 141 |
| <i>Sergej JOHANN, Christoph STRANGFELD, Omid CHAHARDEHINEJAD, Matthias BARTHOLMAI</i>                                                               |     |
| COMPARATIVE RESULTS OF MILLING STRATEGIES IMPLEMENTATION .....                                                                                      | 143 |
| <i>Adrian BUT, Radu CANARACHE</i>                                                                                                                   |     |
| TOW TRUCK FRAME MADE OF HIGH STRENGTH STEEL UNDER CYCLIC LOADING.....                                                                               | 145 |
| <i>Tadeusz SZYMCZAK, Adam BRODECKI, Katarzyna MAKOWSKA, Zbigniew L. KOWALEWSKI</i>                                                                  |     |
| STATIC TESTS ON A RAILWAY VEHICLE BODY BEFORE BOGIE REPLACEMENT.....                                                                                | 147 |
| <i>Nicușor Laurențiu ZAHARIA, Dan Mihail COSTESCU, Elvis-Georgian ZIDARU, Ștefan Dan PASTRAMĂ</i>                                                   |     |
| EXAMINATION OF THE INTERMODAL TRAILER FRAME IN ACCORDANCE WITH UIC 596-5 .....                                                                      | 149 |
| <i>Jacek KARLIŃSKI, Paulina DZIAŁAK</i>                                                                                                             |     |
| STRENGTH OF BRIDGE STRUCTURES: A STUDY .....                                                                                                        | 151 |
| <i>Mihai-Liviu BĂIAȘU, Diana-Ionela SANDU, Marilena MANEA, Andrei-Mădălin GRIGORE</i>                                                               |     |
| 1:60 SCALED EARTHQUAKE TESTING OF THE IMMERSSED TUNNEL OF THE HONG KONG-ZHUHAI-MACAO-BRIDGE .....                                                   | 153 |
| <i>Eva BINDER, Chong LI, Herbert MANG, Yong YUAN, Bernhard PICHLER</i>                                                                              |     |

|                                                                                                                                         |     |
|-----------------------------------------------------------------------------------------------------------------------------------------|-----|
| ASSESSMENT OF GRINDING WHEEL WEAR VIA MONITORING OF PROCESS POWER .....                                                                 | 155 |
| <i>Clemens SCHWAIGER, Matthias HACKSTEINER, Friedrich BLEICHER</i>                                                                      |     |
| MULTI-INTERVAL LOAD CELL WITH STEPLESS AUTOMATIC RANGE TRANSITION ACCORDING THE WEIGHT.....                                             | 157 |
| <i>Martin STOCKMANN, Thomas LEHMANN, Jörn IHLEMANN</i>                                                                                  |     |
| CONTROL OF A SPHERICAL PARALLEL MANIPULATOR WITH THREE DEGREE OF FREEDOM .....                                                          | 159 |
| <i>Dino HÜLLMANN, Harald KOHLHOFF, Jessica ERDMANN, Patrick P. NEUMANN</i>                                                              |     |
| APPLICATION OF AN ELECTRIC CONTACT SENSOR IN MACHINING .....                                                                            | 161 |
| <i>David HAJDU, Daniel BACHRATHY, Tamás INSPERGER, Gabor STEPAN</i>                                                                     |     |
| COMPARISON OF COOLING SIMULATIONS OF INJECTION MOULDING TOOLS CREATED WITH CUTTING MACHINING AND ADDITIVE MANUFACTURING.....            | 163 |
| <i>József Bálint RENKÓ, Dávid Miklós KEMÉNY, József NYIRÓ, Dorina KOVÁCS</i>                                                            |     |
| CUTTING FORCE MEASUREMENT FROM ACCELERATION SENSOR IN MILLING OPERATION.....                                                            | 165 |
| <i>Adam KISS, Daniel BACHRATHY, Gabor STEPAN</i>                                                                                        |     |
| IMPROVED SPINDLE SPEED MEASUREMENT BY ANALOGUE SENSOR.....                                                                              | 167 |
| <i>Daniel BACHRATHY</i>                                                                                                                 |     |
| DESIGN THINKING IN NUMERICAL AND EXPERIMENTAL EVALUATION OF THE STRUCTURE OF CONSTRUCTION ROBOT.....                                    | 169 |
| <i>Damian DERLUKIEWICZ, Jakub ANDRUSZKO</i>                                                                                             |     |
| EXPERIMENTAL AND NUMERICAL EVALUATION OF WAVE IMPACT STRESS ON A COMPOSITE BOAT HULL.....                                               | 171 |
| <i>Adrian CARAMATESCU, Costel Iulian MOCANU</i>                                                                                         |     |
| OPTICAL STUDY FOR SPRINGBACK PREDICTION, THICKNESS REDUCTION AND FORCES VARIATIONS ON SINGLE POINT INCREMENTAL FORMING .....            | 173 |
| <i>Nicolae ROȘCA, Valentin OLEKSIK, Adrian PASCU, Mihaela OLEKSIK, Eugen AVRIGEAN</i>                                                   |     |
| DYNAMIC TESTS PERFORMED ON FLAT FREIGHT WAGON.....                                                                                      | 175 |
| <i>Nicușor Laurențiu ZAHARIA, Alexandru Ionuț PĂTRAȘCU, Dan Mihail COSTESCU</i>                                                         |     |
| EXPERIMENTAL STRESS ANALYSIS OF A RAILWAY CAR USING A DEDICATED TEST RIG .....                                                          | 177 |
| <i>Elvis-Georgian ZIDARU, Alexandru Ionuț PĂTRAȘCU, Nicușor Laurențiu ZAHARIA, Anton HADĂR</i>                                          |     |
| EXPERIMENTAL STRESS ANALYSIS ON RAILWAY VEHICLE BODY .....                                                                              | 179 |
| <i>Alexandru Ionuț PĂTRAȘCU, Nicușor Laurențiu ZAHARIA, Elvis-Georgian ZIDARU, Anton HADĂR</i>                                          |     |
| EXPERIMENTAL STUDY OF THE DYNAMIC RESPONSE OF A CABLE UNDER WIND FLOW .....                                                             | 181 |
| <i>Vlad Daniel URDAREANU, Ionuț Radu RĂCĂNEL, Mircea DEGERATU, Costin Ioan COSOIU, Cristian Lucian GHINDEA, Radu Iuliu CRUCIAT</i>      |     |
| KEY FACTORS TOWARDS A HIGH QUALITY ADDITIVE MANUFACTURING (3D PRINTING) PROCESS WITH ABS MATERIAL .....                                 | 183 |
| <i>Radu Emanuil PETRUSE, Samuel PUȘCAȘU, Adrian PASCU, Ioan BONDREA</i>                                                                 |     |
| STUDY REGARDING THE GEOMETRICAL DEVIATIONS FOR 3D PRINTED FUNCTIONAL PARTS.....                                                         | 185 |
| <i>Samuel PUȘCAȘU, Radu Emanuil PETRUSE, Adrian PASCU, Ioan BONDREA</i>                                                                 |     |
| EXPERIMENTAL STUDIES OF A BRIDGE MADE FROM REINFORCED CONCRETE BEAMS - BRIDGE OF TRANSILVANIA HIGHWAY .....                             | 187 |
| <i>Polidor BRATU, Ovidiu VASILE, Cătălin ZAHARIA</i>                                                                                    |     |
| EVALUATION OF ELASTOMERIC STRUCTURAL BEARINGS PERFORMANCE BASED ON NATURAL SCALE TESTS.....                                             | 189 |
| <i>Polidor BRATU, Ovidiu VASILE, Sorin POPA</i>                                                                                         |     |
| ASSESSMENT OF SOIL STRENGTH AND DEFORMABILITY PARAMETERS IN LABORATORY CONDITIONS.....                                                  | 191 |
| <i>Cornelia-Florentina DOBRESCU, Oana TONCIU</i>                                                                                        |     |
| SPECIALISED TESTS FOR THE ASSESSMENT AND CLASSIFICATION OF SAFES FOR THE VALUES PROTECTION, FROM THE SECURITY IN USE POINT OF VIEW..... | 193 |
| <i>Cristinel SEBE, Cătălin ZAHARIA, Sorin POPA</i>                                                                                      |     |

|                                                           |     |
|-----------------------------------------------------------|-----|
| HARDWARE IN THE LOOP SIMULATION FOR HOMING MISSILES ..... | 195 |
| <i>Ivana TODIĆ, Vladimir KUZMANOVIĆ</i>                   |     |

## SECTION V: NUMERICAL METHODS

|                                                           |     |
|-----------------------------------------------------------|-----|
| SEISMIC CALCULATION FOR GENERATOR PROTECTION PANEL.....   | 199 |
| <i>Paweł MAŚLAK, Jerzy CZMOCHOWSKI, Tadeusz SMOLNICKI</i> |     |

|                                                                                            |     |
|--------------------------------------------------------------------------------------------|-----|
| ANISOTROPY OF PERIODIC MICROSTRUCTURE IN MODELS OF UNIDIRECTIONAL COMPOSITE MATERIALS..... | 201 |
| <i>Robert ZEMČÍK, Hana ZEMČÍK, Tomáš KROUPA</i>                                            |     |

|                                                                                     |     |
|-------------------------------------------------------------------------------------|-----|
| ESTIMATIONS OF THE SHIP HULL OSCILLATIONS .....                                     | 203 |
| <i>Dumitru LUPAŞCU, Ionel CHIRICĂ, Elena-Felicia BEZNEA, Mihaela BARCAN (SAVIN)</i> |     |

|                                                                                    |     |
|------------------------------------------------------------------------------------|-----|
| SHIPS'S HULL STRUCTURAL SAFETY ESTIMATION.....                                     | 205 |
| <i>Dumitru LUPAŞCU, Ionel CHIRICĂ, Liviu-Dan STOICESCU, Mihaela BARCAN (SAVIN)</i> |     |

|                                                                                         |     |
|-----------------------------------------------------------------------------------------|-----|
| FINITE ELEMENT STUDY ON THE STRESS STATE ACROSS THE THICKNESS OF A CURVED LAMINATE..... | 207 |
| <i>Marius Nicolae BABA</i>                                                              |     |

|                   |     |
|-------------------|-----|
| AUTHOR INDEX..... | 209 |
|-------------------|-----|

## FOREWORD

Each year since 1984, the Danubia-Adria Symposium on Advances in Experimental Mechanics brings together internationally recognized experts and young researchers in an effort to exchange ideas in different topics having as common link Experimental Mechanics. Since the beginning, this Symposium was a platform for establishing connections between different research groups trying to establish future scientific collaboration and an agora where faculty members, students, scientists, researchers, engineers and industrial experts present and discuss the status and impact of modern technology and development in the field of experimental mechanics.

In 2018, the 35th Danubia Adria Symposium on Advances in Experimental Mechanics is to be held between September 25 – 28 in the wonderful mountain resort Sinaia, Romania.

The Book of Abstracts of the proceedings includes 97 papers, which have been selected and accepted after peer-review by the Scientific Committee, formed by the delegates of the eleven DAS member countries – Austria, Croatia, Czech Republic, Germany, Hungary, Italy, Poland, Romania, Serbia, Slovakia, and Slovenia.

The papers are organized in five sections:

- Structural analysis
- Materials characterization and testing
- Biomechanics
- Practical applications/case-studies/instrumentation
- Numerical methods

The present volume emphasizes the actual trends which are given to the development of methods and models that account for and integrate physical phenomena taking place on multiple scales. These include complex processes such as the evolution of the meso- and micro-structure of the material during loading, and if these phenomena could be accounted for, then their influence on the macroscopic constitutive behavior of materials could be also predicted, thus being able to create new, exceptional materials. While this perspective is fascinating, the current state-of-the-art falls short from attaining this goal. The better understanding of phenomena where large populations of defects - such as dislocations - interact in highly non-linear ways and the consideration of large deformations can lead to the creation of improved constitutive models of the materials' behavior to be observed macroscopically.

Finally, it is important to underline that, with the 2018 Romanian edition of the Symposium, the Danubia Adria Society on Experimental Methods proudly announces its 35th anniversary!

We wish all the participants to have a high level of scientific discussions and to enjoy the beauty of the Romanian mountains.

The Editors,

Professor Ștefan Dan Pastramă

Professor Dan Mihai Constantinescu



## **SECTION I: STRUCTURAL ANALYSIS**



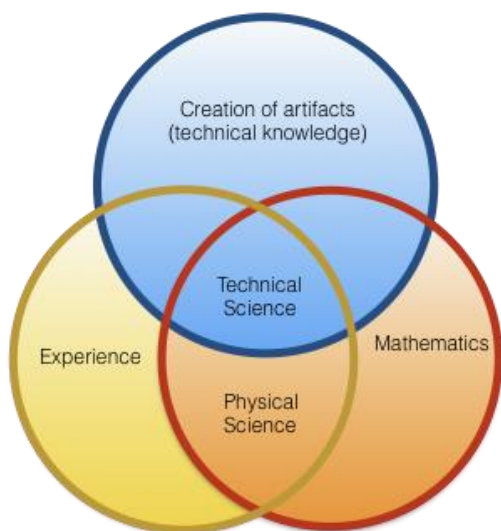
# EXPERIMENTING FOR KNOWING: EVOLUTION OF A CONCEPT IN TECHNICAL SCIENCE

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## 1. Introduction

Physical Science lies at the intersection of experience (observations) and mathematics. Technical Science, on the other hand, derives from the interaction between Physics and technical knowledge (Fig. 1).



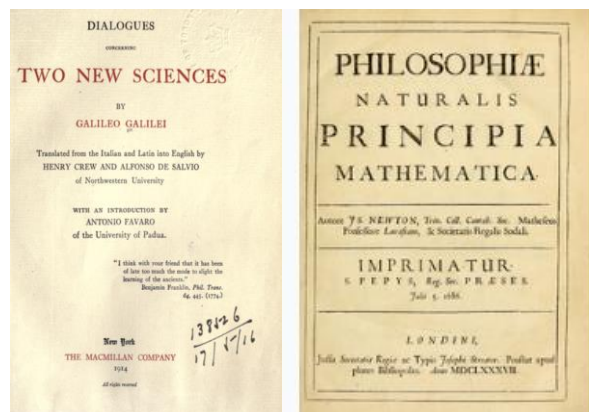
**Fig. 1:** Technical Science at the intersection of three areas: observing, measuring and making

With regard to this, in Albert Einstein's *Autobiography* we read:

*“There [in ETH] I had excellent teachers (for example, Hurwitz, Minkowski), so that I really could have gotten a sound mathematical education. However, I worked most of the time in the physical laboratory, fascinated by the direct contact with experience. ...”*

## 2. The watershed between tests and experiments

The watershed between tests and experiments are Galileo Galilei's books, in particular the *Dialogues Concerning Two New Sciences* (1638): Statics. Then, Isaac Newton's *Philosophiæ Naturalis*



**Fig. 2:** *Two New Sciences* and *Principia Mathematica*  
title pages

*Principia Mathematica* (1686-1687). In Einstein's words, "...the first in each pair intuitively captured the possible relationships, while the second formulated them exactly, and applied them quantitatively."

### 3. A Test is not an Experiment

Test and experiment have two different meanings. A Test is an activity undertaken on a system in order to determine whether or not this meets the expectations under real operating conditions. If it does not, appropriate modifications must be introduced. Experiments, conversely, allow to check which variables (called factors) have an influence (effect) on the response of the system, i.e. for which variable values the system gives an optimal response. A Test is used to verify an intuition, while an Experiment to understand a process.

#### 4. Physical Science needs Technical Science

That Physical Science needs Technical Science is also clear in the following quotation from Burton Richter's Autobiography, Nobel Prize in Physics, 1976:

*"Modern science is fast-moving and no laboratory can exist for long with a program based on **old facilities**. Innovation and renewal are required to keep a laboratory on the frontiers of science and only if it remains on the frontiers will it have a long-term future. ... It is important that these resources be made available **even in times of tight budgets**. Starving the future to feed the present is a mistake - it leads to obsolescence and stagnation..."* Another example of this fruitful interaction are the CMS experiments at CERN's Large Hadron Collider to validate Peter Higgs' Standard Theory.

#### 5. Before Galilei

Experimenting is Testing without the knowledge of any laws of Statics, to understand only the thrusts directions, but not the values nor the stresses. Experimental models provide correctly the directions, but not the entity.

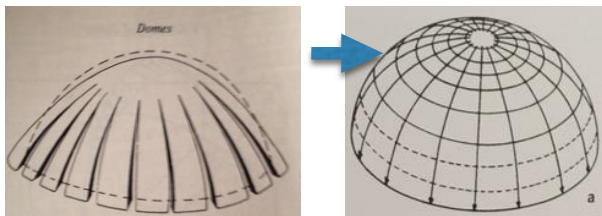


Fig. 3: Thrust and stresses in a dome.

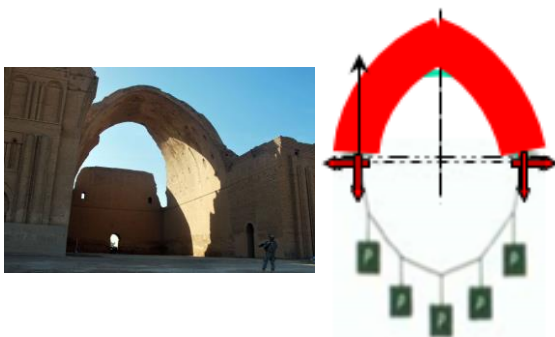


Fig. 4: The analogy of the chain in Ctesiphonts arch (3rd-6th century Sasanian Persian monument,) probably discovered before R. Hook (1629).

The medieval view of the masonry structure was based on geometry: the safety of the structure was related to the stability (proportion of parts) and not to stress: it is wrong but acceptable, provided the stresses are low.

Galileo gave the first demonstration that a scale model will always be misleadingly strong.

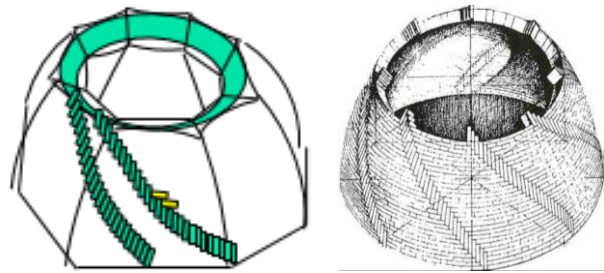


Fig. 5: Technique without science: Building of a big dome without scaffolding

#### 6. After Galilei: Mathematics from Servant to Mistress

*"The most fascinating subject at the time that I was a student was Maxwell's theory. What made this theory appear revolutionary was the transition from forces at a distance to fields as fundamental variables. ... the Faraday-Maxwell pair had a similar resemblance to the Galileo-Newton pair ..."* (A. Einstein).

Of Maxwell, Hertz said: *"Die Gründung der Maxwellsche Theorie ist das System der Maxwellschen Gleichungen"* [The foundation of Maxwell's theory is the system of Maxwell's equations].

Einstein's revolution consists in the transformation of cosmology into a real science, by providing a set of mathematical equations, the solutions of which (there exists an infinite number) describe all together all universes that are possible. These mathematical descriptions provide predictions that astronomers control through satellites and telescopes.

Quoting once more from Einstein's autobiography, "*Beyond this however, my interest in the knowledge of nature was also unqualifiedly stronger; and it was unclear to me, as a student, that the approach to a more profound knowledge of the basic principles of physics is tied up with the most intricate mathematical methods. This dawned upon me only gradually after years of independent scientific work.*"

## 7. Experimenting in the Industrial Revolution: The role of the Phenomenological Theories

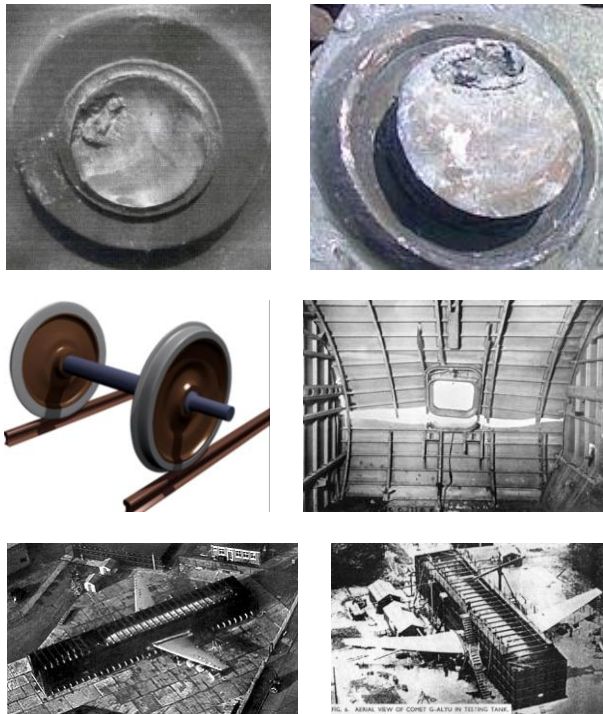


Fig.6: Examples of in-field experiments based on phenomenological theories

Phenomenological theories are theories of practical interest, not entirely derived from the great theories of science. The first collection of data should be conducted in-field in order to identify all the variables that influence the phenomenon.

After this, the laboratory is used to quantify the influence of the main factors (and interactions) that have a real influence.

## 8. New Mathematical Models

To describe the stress due to a crack, the functions of complex variables (functions of complex numbers) are necessary. Fracture mechanics is a phenomenological science built on:

- New observational studies
- New mathematics

## 9. Experiments Variability

Vilfredo Pareto (1848–1923) and sir Ronald Fisher (1890–1962) studied the variability of experimental data and the influence of factors on output variables.

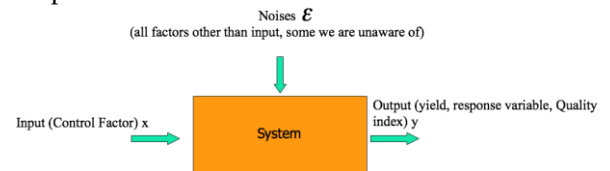


Fig. 7: The diagram for the Design of Experiment

The Design of Experiments (DOE) is developed. The pairs mentioned so far (Galilei-Newton, Faraday-Maxwell, Pareto-Fisher) can be completed with the names of technical scientists, who have developed practical applications of the theory, (creation of artifacts): for example, Eiffel, Tesla, Marconi, ... Taguchi.

## 10. Interpretation of Signal /Noise

The concept of robustness proposed by Taguchi can be interpreted beyond its technical meaning as the:

- Ratio of relevant to irrelevant information
- Minimization of noise by removing unnecessary elements
- Measurement of how much useful information there is in a system, (e.g. the Internet), as a proportion of the entire contents

## 11. Simulation

In the computer era, experimenting takes on a new form. A numerical simulation is a calculation that is run on a computer following a program that implements a mathematical model for a physical system. The computer allows to quickly process the

model and get “experimental” results cheaper and faster than in a physical model or under real-life conditions.

## 12. New task of Experimentation

- Use the data collected in a large number of experiments and draw new conclusions, (which are invisible in cases of limited experimentation)
- Predict system performance in conditions not directly tested.
- Compare different systems.
- Manage your lab data centrally & allow collaboration of more units.
- The numerical simulation is valid when the physics of the phenomenon is fully known.
- Vice versa, the physical experiment is valid even if the physics of the phenomenon is not fully known.

## 13. Conclusions

Experimenting in structural mechanics is a concept in constant evolution. Probably, at the beginning of human experience to make an experiment meant only adapting and improving artifacts to adjust to new environment conditions (strong earthquakes, materials aging etc.). Over the centuries it has become the only way for Technical Science to advance. Technical Science develops and proceeds independently from Physical Science and the two paths crossed only on specific historical occasions. The seventeenth century was certainly the time when the most favourable conjuncture is reached. Science and Technology meet in the visions of Galileo and Newton, with the logical support of mathematics.

In Galileo, this meeting takes place in an arsenal where construction is practiced and in the observation of the stars. In Newton, in the observation of the movement of celestial bodies. Newton develops a new mathematics, the differential calculus to correctly interpret motion. The mathematical model becomes very important, even in a mechanical vision of the cosmos. After that, mathematical modelling becomes not only necessary but also constitutive. However, it is still the experimental basis offered by Faraday that allows Maxwell’s full mathematical modelling. Finally, Einstein interprets the cosmos only through a mathematical model.

What about Technical Science? Several theories have developed, all connected with a given physical phenomenon, which cannot be derived from a single general theory, but live together in a sort of a patchwork theory. They do not have to be contradictory but often ignore one another as the theory of Material Fatigue, of Fracture Mechanics, of Aging of Materials, etc. They cover several aspects without a single thought. Nevertheless, safety, reliability and other social requirements keep improving, until proven otherwise. In this case Technical Sciences must find the remedy that society requires.

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## EXPERIMENTAL AND NUMERICAL STUDY ON THE BENDING STRENGTH OF A T-CORE SANDWICH PANEL

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### 1. Introduction

The design of large structures requires conditions difficult to be fully assured: low weight and cost, high strength, stiffness and stability. A structural component that responds to such requirements is a metallic or composite sandwich panel. A sandwich panel consists of a lightweight core that is fixed between two high resistant plates (the skins). Depending on the involved materials, the components are assembled with adhesive, by welding or with mechanical fasteners. The cores of high strength panels can be from aluminum or steel, in the form of a corrugated board or consisting of parallel profiles with section I, C, Z, O, V [1-6]. In the case of medium-strength panels, cores of polymeric foams or honeycombs of aluminum foil or impregnated paper are preferred.

In this work, a sandwich panel with the core consisting of parallel T profiles disposed alternatively, standing and overturned (Fig. 1), was analyzed experimentally and by finite element analysis (FEA). The panel components, galvanized steel profiles and aluminum faces, were assembled with structural adhesive AW 106 (from Huntsman).

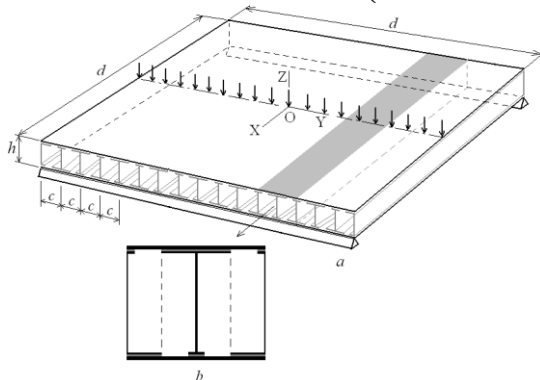


Fig. 1. Geometry of the sandwich panel.

Specimens of beam type can be cut from the panel and tested at three-point bending. Numerical analyses with finite elements have been done both for evaluating the bending response of an

entire panel [5] and for this test. FEAs provided additional data regarding the loading and failure of the panel components (adhesive, core, faces).

### 2. Three points bending tests

FEA and analytical calculation can be done by partial modeling of the panel, considering a finite sheet (equivalent beam), which is shaded in Fig. 1. The cross section of this equivalent beam contains two halves of T profile. For technological reasons the specimen was different, containing three whole T profiles (Fig. 2) and having a smaller length than the entire panel. Therefore, it was necessary to simulate the test for this geometry by using FEA.

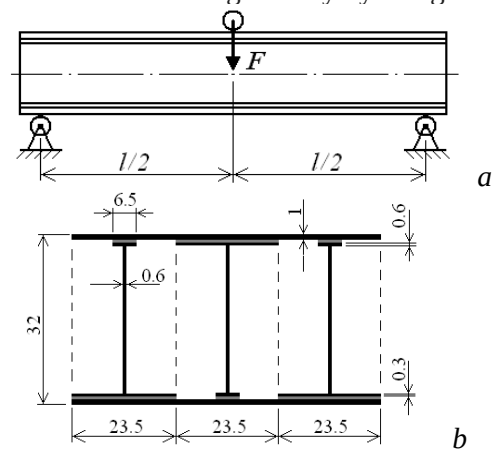


Fig. 2. Loading scheme (a) and the cross section of the specimen (b).

A universal testing machine Lloyd Instruments LRXplus with the nominal force of 5 kN and a special device (Fig. 3) were used to support and load the specimens. The span between the supports was  $l = 240$  mm and the piece used to apply the load were a steel roll of 10 mm in diameter (Fig. 3). Diagrams that emphasize the variation of maximum deflection as a function of applied force were obtained after testing (Fig. 4).

The first peak in the recording is due to local failure of adhesive at the top of the middle section of the specimen and the second peak is associated

with the local buckling of the core profiles.



Fig. 3. A specimen under bending.

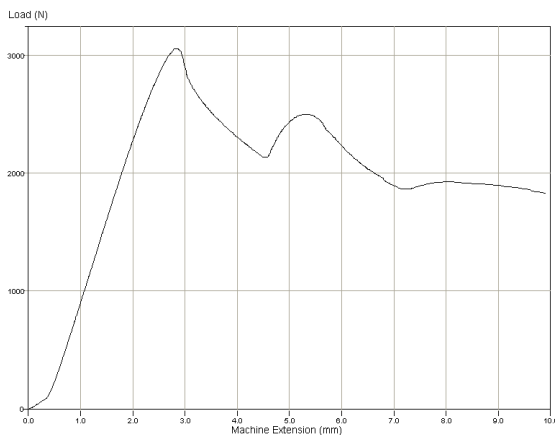


Fig. 4. A load-deflection curve.

### 3. Results of finite element analyses

A linear elastic FEA was performed by using ANSYS [7]. Taking into account the symmetry, a model with brick-type finite elements was generated for a quarter of the specimen and the appropriate boundary conditions were imposed in the section planes. The elastic parameters of materials (Young's modulus and Poisson's ratio) were:  $E_f = 69$  GPa,  $\nu_f = 0.33$  (for the aluminum alloy),  $E_m = 210$  GPa,  $\nu_m = 0.3$  (for the steel),  $E_a = 1.8$  GPa,  $\nu_a = 0.44$  (for the adhesive).

The loading was produced by imposing a displacement of one millimeter on the central cylinder (Fig. 2). By summing the reactions from the supports it was found that this displacement corresponds to a force  $F \approx 2800$  N.

The experimental diagram from Fig. 4 shows a nearly double displacement to this force due to the progressive settlement of the specimen heads on the supports and due to the local plastic deformation in the center of the upper face.

Finite element analysis was linear elastic, so it did not reveal the local stability loss of the profiles, but it clearly showed that large equivalent stresses occur in the central cross section of the specimen: 30 MPa in the adhesive, 180 MPa in the faces, 500 MPa in the profiles.

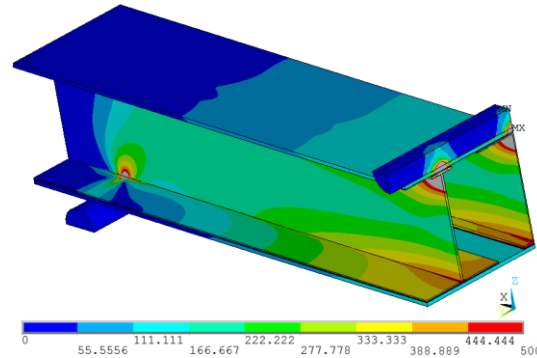


Fig. 5. Equivalent stresses in a quarter of the specimen.

### 4. Conclusions

The FEAs and tests provide complementary information on quantitative and qualitative aspects of the behavior of sandwich structures and samples taken out from them. Thus, based on the information obtained, designers can improve their initial solutions.

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# IMPACT ON A MICRO-REINFORCED UHPC: EXPERIMENTAL STUDIES VS. NUMERICAL MODELING

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## 1. Introduction

For security reasons, the development of barrier elements to restrain terroristic attacks is nowadays of high public interest. Within a joint research project, specific mobile barriers are developed for public space installations which, on the one hand, will be architecturally pleasing and on the other hand are highly effective in dissipating energy from vehicle impacts or other forms of attack.

This contribution describes experimental and numerical investigations for developing a thin-shelled, modular, mobile element system. As material, the micro-reinforced ultra-high-performance ductile concrete (DUCON<sup>®</sup>) is used, which possesses a high impact resistance by a high energy absorbing capacity.

As a first step, material parameters of this unique concrete were experimentally determined (quasi-static, dynamic) to adapt the material model within the numeric analysis on the base of these data. Afterwards, for validation of the numerical model quasi-static and high-velocity impact tests were performed on plate-like structures. Photogrammetric surveys were executed to receive an exact image of the behavior of the test specimen.

Finally, a suitable geometry of barrier elements will be designed, which provides a maximum of effectivity against impact by a minimum of weight and a maximum of mobility.

## 2. Methods

### 2.1 Experiments

To determine compressive strength, splitting tensile strength and Young's modulus, quasi-static experiments were conducted on samples of small cylinders (60 x 120 mm) of the high performance

DUCON<sup>®</sup> concrete (max. size of aggregate 2 mm [1]). The tests were conducted using the MTS/1 MN machine at the BAM laboratory (Fig. 1a).

For validation of a numerical model, 6 further quasi-static load tests were carried out on micro-reinforced DUCON<sup>®</sup> slabs of 1.52 m x 1.52 m and with a thickness between 2.5 cm and 15 cm (Fig. 1d). For those tests the unique impact test device at the BAM lab was utilized (Fig. 1b). The slabs were seated on four spheres at their corners.

Experiments of dynamic loading are also being carried out on cylinders and on 6 micro-reinforced slabs (equal dimensions). For all slab-tests an impact velocity of  $v = 3.5$  m/s and a resulting force of  $F = 400$  kN was defined based on the characteristic curve of the 1000 kN cylinder. For the sample cylinder-tests velocities of  $v = 4, 40, 400, 4000$  mm/s and forces of  $F = 1000, 995, 980$  and  $200$  kN were chosen (Fig. 1c).

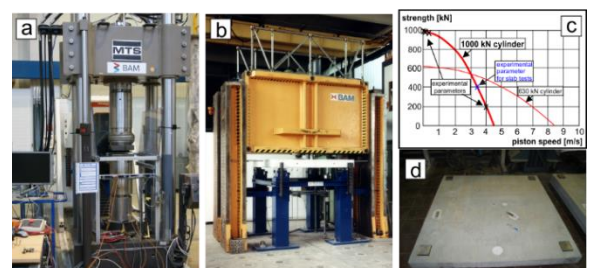


Fig. 1. a): MTS/1MN device for quasi-static experiments, b): impact testing device for dynamic experiments with embedded concrete slab, c): characteristic curve of the two possible cylinders of b) and exp. Parameters, d): used concrete slab with the 4 bearing plates, 2 applied strain gauges and positions for inductive displacement transducer (light grey dots).

### 2.2 Numerical Modeling/Finite Element Simulation.

The simulation was performed using the commercial software ANSYS<sup>®</sup> AUTODYN.

Within the numerical analysis a cylindrical specimen of DUCON<sup>®</sup> concrete was pressed between two structural steel bodies. Since DUCON<sup>®</sup> concrete is very homogeneous it was assumed that the material behaves isotropic. In the elastic range, the Young's modulus and the Poisson's ratio were determined experimentally. The Mohr-Coulomb failure criterion was used for the plastic behavior. Therefore, also the material parameter internal friction angle and cohesion needed to be determined experimentally by, in our case, a splitting tensile test and a compression strength test. The bearing was specified as frictionless. The specimen was loaded in a displacement controlled manner.

### 3. Results

#### 3.1 Experiments-quasi-static and dynamic

The quasi-static experiments on the cylinders result in mean compressive strength of  $f_c = 118 \text{ N/mm}^2$ , a splitting tensile strength of  $f_t = 5.7 \text{ N/mm}^2$  and an Young's modulus of  $E = 38900 \text{ N/mm}^2$ . The first static test at the slab with thickness  $h = 120 \text{ mm}$  (P 120) showed an ultimate load of  $F = 380 \text{ kN}$ . The maximum deflection of this slab was  $d = 29 \text{ mm}$ . P 80 showed a maximum load of  $F = 182 \text{ kN}$  and a deflection of  $d = 54 \text{ mm}$  (Figs. 2, 3). The evaluation of the photogrammetric data (e.g. strain, displacement, crack propagation, compare Fig. 3b) is in progress.

In relation to the quasi-static tests, the first dynamic test (P 100) showed a higher maximum ultimate load ( $F = 430 \text{ kN}$ ) and a larger bending deflection ( $d = 72 \text{ mm}$ ) (Figs. 2, 3).



Fig. 2. Comparison of the maximal bending of the concrete slabs at attending the maximal ultimate load. a): dynamic test P 100, b): quasi-static test P 120, c): quasi-static test P 80

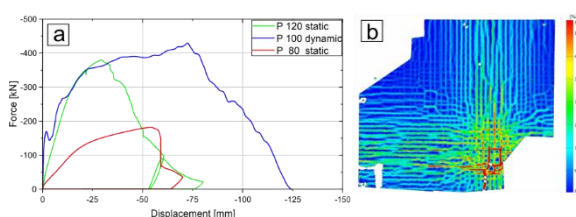


Fig. 3. a): Force vs. displacement for three slab tests; note that P 100 (blue line) was a dynamic test, b) photogrammetric survey of crack propagation (P 80)

#### 3.2 Numerical Modeling-quasi-static tests

The results of the numerical elastic compression strength tests are in good agreement with experimentally determined Young's modulus (Figs. 4a and 4b). The numerically determined Young's modulus is  $E = 38315 \text{ N/mm}^2$ .

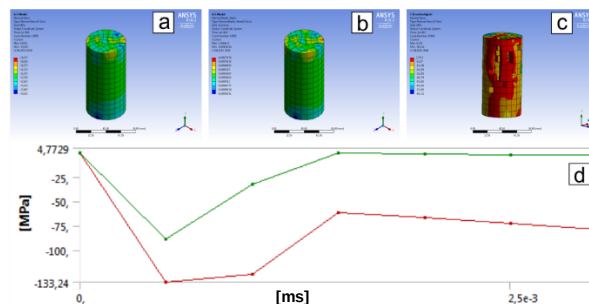


Fig. 4. FE modeling of stress (a), strain (b) and failure (c). d): progression of normal stress vs. time

The results of the plastic compression strength tests are shown in Fig. 4c. It can be seen, that cracks occurred at a strength of  $\sigma \sim 133 \text{ N/mm}^2$  (Fig. 4d). With  $18 \text{ N/mm}^2$  it is slightly higher than expected.

### 4. Conclusion

The described first tests clearly show that the ductility increases the thinner the slab is and further, the dynamic test shows that the material and so the structural resistance increases with the loading velocity. By now a good agreement between experimental and numerical analysis could be received. The numerical results will be further improved by validating the FE models on the base of the experimental results.

Further quasi-static and dynamic tests on slabs are in progress as well as dynamic tests on cylindric specimen. With the complete set of experimentally determined material parameters the FE model will be validated and finally, an optimal geometry for the barrier elements will be investigated based on numerical modeling. Ultimately, the so determined barrier structure will be tested in a complex experimental setup by impact loading.

### Acknowledgements

This work was funded by AIF: DUCON<sup>®</sup>-Mobil safety Wall; support code ZF4044214HF7.

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# A METHOD FOR STRAIN ANALYSES OF SURFACES WITH CURVED BOUNDARIES BASED ON MEASURED DISPLACEMENT FIELDS

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## 1. Introduction

The determination of field information regarding displacement and strain is frequently performed in stress analyses of structures and in material mechanics investigations. Therefore, effective and high-performance methods are required in different measurement ranges. In the small deformation range, Electronic Speckle Pattern Interferometry (ESPI) using phase shifting technique is well suited to determine displacements and strains very precisely caused by the high resolution and sensitivity of the method. The strains can be obtained using the evaluation of approximated displacement functions. In this contribution, this method is performed with an additional parametrization of a surface section with curved part to improve the evaluation near the edge.

## 2. Experimental technique

In Fig. 1 the geometry of the investigated epoxy specimen and the load case is given. The notched cantilever beam is loaded by a displacement  $u_{punch}$ . For the measurement of the in-plane displacement fields, ESPI with temporal phase shifting technique was used. This method includes the calculation of the phase angles of resulting intensities due to interference. The phase angle difference  $\Delta\phi$  is based on two conditions (unloaded, loaded) and determined for the directions  $x$  and  $y$ . Details to the principle and application examples are given in [1].

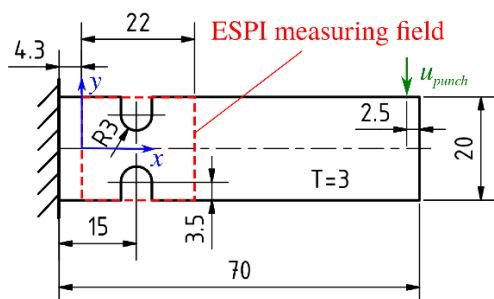


Fig. 1. Specimen geometry, load case, measuring field.

In the presented investigations, a proprietary developed 2D ESPI device was used. Phase shifting technique was performed by means of mirrors with piezo actuator.

## 3. Evaluation method

The primary results are fields of the phase angle differences with discontinuities caused by the range  $[-\pi, \pi]$ . Thus, phase unwrapping is required to determine the continuous displacement fields  $u_x$  and  $u_y$ . The experimental displacement data was approximated by cubic B-spline surfaces using Matlab [2,3]. For the performed approximation, gridded data of the displacements is necessary. By means of the derivatives of the displacement functions, the strains are determined. Performing a global analysis of the surface in case of a notched surface geometry, the gap of displacement data in the notch area has to be filled by interpolation. The advantage of this variant is the capability to analyze the whole measuring field. However, inaccuracies at the edges of the notched contour occur.

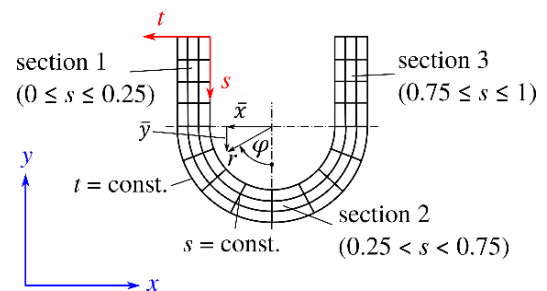


Fig. 2. Parametrization of the notch-surrounding section.

To evaluate the section around the upper notch with improved accuracy, a method performing surface parametrization was used, see Fig. 2. The notch-surrounding surface is divided in three segments (two rectangles (sections 1 and 3) and one annulus (section 2)). Furthermore, to achieve a compact consideration, additional coordinates  $\bar{x}(x)$  and  $\bar{y}(y)$  are introduced. The parameters  $s$  and  $t$  are

normalized ( $s, t \in [0,1]$ ) and represent functions of the coordinates, that is  $s(y)$  and  $t(x)$  in the rectangular sections as well as  $s(\bar{x}, \bar{y})$  and  $t(\bar{x}, \bar{y})$  in the annular section. The relations between the parameters and the coordinates are obtained by the given geometrical conditions. Using B-spline approximation with Matlab including parametrization, the functions  $u_x(s, t)$  and  $u_y(s, t)$  are provided. The resulting in-plane strains are calculated by means of the following relations:

$$\varepsilon_{xx} = \frac{\partial u_x}{\partial x}, \quad \varepsilon_{yy} = \frac{\partial u_y}{\partial y}, \quad \varepsilon_{xy} = \frac{1}{2} \left( \frac{\partial u_x}{\partial y} + \frac{\partial u_y}{\partial x} \right) \quad (1)$$

Exemplarily, the strain  $\varepsilon_{xx}$  using the derivatives of the displacement  $u_x$  with respect to the parameters and using the derivatives of the parameters with respect to the coordinate  $x$  is given by:

Rectangular sections:

$$\varepsilon_{xx} = \frac{\partial u_x}{\partial x} = \frac{\partial u_x}{\partial t} \frac{dt}{dx} \quad (2)$$

Annulus section:

$$\varepsilon_{xx} = \frac{\partial u_x}{\partial x} = \frac{\partial u_x}{\partial s} \frac{\partial s}{\partial \bar{x}} \frac{d\bar{x}}{dx} + \frac{\partial u_x}{\partial t} \frac{\partial t}{\partial \bar{x}} \frac{d\bar{x}}{dx} \quad (3)$$

In the transition zones of the sections, significant changes of the displacement distribution depending on parameters  $s$  and  $t$  can occur. Hence, smoothing of the strain values improves the results in these zones.

## 4. Results

The results are demonstrated using exemplarily the evaluation in x-direction. In Fig. 3 the phase map is given for a punch displacement of  $u_{punch} = 20 \mu\text{m}$ . The resulting displacement field for the x-direction using phase unwrapping is shown in Fig. 4. Here, the displacement difference  $\Delta u_x$  referring to a value  $u_x(x = 0.5 \text{ mm}, y = 0)$  (not determined) is demonstrated.

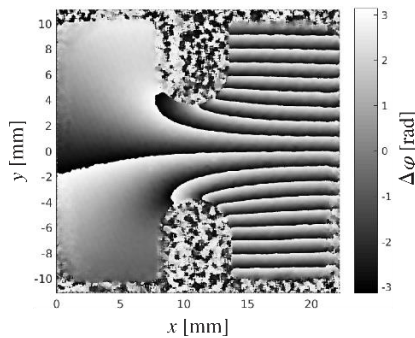


Fig. 3. ESPI phase map, evaluation in x-direction.

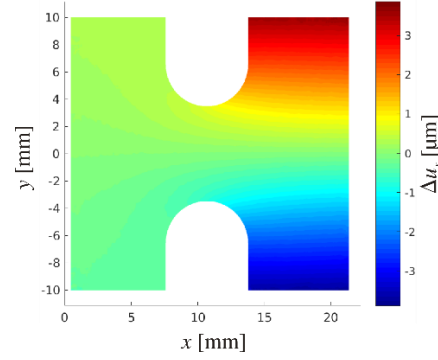


Fig. 4. ESPI displacement field  $\Delta u_x(x, y)$ .

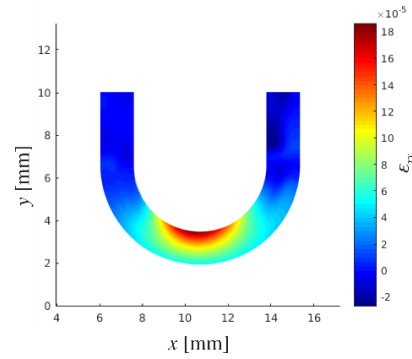


Fig. 5. Resulting strain field  $\varepsilon_{xx}(x, y)$  in the notch-surrounding section.

Using the described parametrization, the surrounding section of the notch is analyzed. The strain  $\varepsilon_{xx}$  calculated by means of Eq. (2) and Eq. (3) using the approximated B-spline function  $u_x(s, t)$  in the parametrized section is shown in Fig. 5. The maximum strain is detected in the middle section of the notch boundary. Note, that the general level of the occurring strains is very small.

## 5. Conclusions

The parametrization of the notch-surrounding area enables a precise determination of the strain. The expected gradients of the strain field  $\varepsilon_{xx}$  at the notch root are represented very well. Furthermore, the described procedure can be adapted for other geometries and for further field measuring methods (e.g. digital image correlation).

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# COMPARISON OF FRACTURE PROPERTIES OF STEEL PIPES

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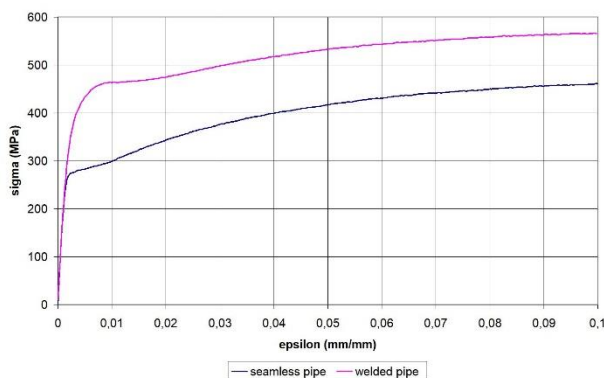
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## 1. Introduction

A research of mechanical and fracture properties of seamless pipe Ø 525/8.5 mm from CSN 413126 steel and axially welded pipe Ø 508/6.3 mm from L360NE steel pointed on the dominance of the latter pipe in basic tensile properties, notch toughness and fracture toughness. Prevailing properties of the welded pipe are due to extending the microalloying of the steel by V, Nb, and Ti with a lower content of S and P and also by a higher degree of working the material during rolling sheet before winding the pipe.

## 2. Comparison of properties of the steels

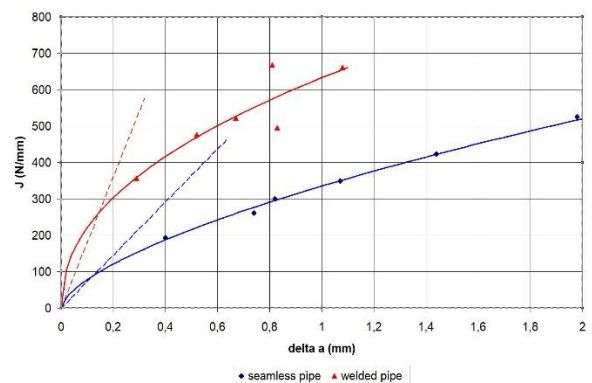
Stress – strain properties were obtained on flat specimens in the hoop direction. Before machining the specimens the curved segments were press straightened. The results are shown in the **Fig. 1**.



**Fig. 1.** Stress-strain curves

Strength parameters for steel 13126 are less than for steel L360NE but, however, the gradient of the hardening above the yield stress is higher, so that for reaching a certain strain in the elasto-plastic region a higher stress is required than for steel L360NE. This is exhibited by a smaller magnitude of the Ramberg-Osgood exponent.

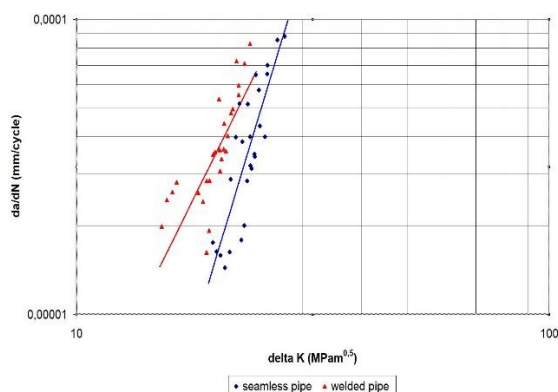
Notch toughness KCV was tested at 0°C using seven charpy specimens from each steel. The thickness of specimens from steel 13126 was ~ 5,3 mm and that of specimens from L360 NE steel was ~ 4,1 mm. Impact energy KV was measured on pendulum impact testing machine with capacity 300 J and then it was transferred to impact notch toughness KCV. Higher magnitudes of KCV were found for L360NE steel (124.5 J/cm<sup>2</sup>) in contrast to 101.3 J/cm<sup>2</sup> for 13126 steel. The significance of the notch toughness KCV consists in principle in two aspects: a) although the notch toughness impact test is a dynamic test its results can be correlated with fracture toughness in terms of the stress intensity factor  $K_c$  which is a static characteristics. This correlation is possible, providing the results of the notch toughness are concerned with the upper plateau of the transition curve [1], b) the results of notch toughness tests on Charpy specimens with a 2/3 thickness make it possible to qualify whether a running crack in the wall of a pressure pipeline would stop in a short distance ( length of a single pipe) or whether it will continue to run in the pipeline wall to a long distance.



**Fig. 2.** R curves

Fracture toughness of the steels was investigated on the basis of the J integral ([2]) with the use of CT specimens. The results are presented on **Fig. 2**. The R curve for L360NE steel lies above that for 13126 steel, indicating thus much better fracture properties. Evaluation of the R curves yielded the following fracture parameters for 13126/L360NE steels, respectively:  $J_{in} = 87/262$  N/mm;  $J_{0.2} = 214/436$  N/mm. It can be seen that  $J_{in}$  for L360NE steel is about three times higher than that for 13126 steel and  $J_{0.2}$  is approximately twice that for steel 13126.

The kinetics of growth of fatigue cracks was investigated during pre-cracking of CT specimens. The results in the form of Paris law dependence are presented on **Fig. 3**.



**Fig. 3.** Paris law dependences

In the range of effective values of the stress intensity factor  $K$  (19-27  $\text{MPa m}^{0.5}$  for 13126 steel and 15-23  $\text{MPa m}^{0.5}$  for L360NE steel) the Paris law dependence for L360NE lies above that for 13126 steel indicating thus a higher fatigue crack growth rate for the steel L360NE.

### 3. Effect of properties of the steels on the integrity of pipes

From the viewpoint of integrity of pipelines the most important property is the fracture toughness. The critical state of a pipeline containing crack comes when the applied J integral at the tip of a crack reaches the fracture toughness identified with e.g. parameter  $J_{0.2}$ . For the pipes investigated it is  $J_{0.2} = 214$  N/mm for 13126 steel and  $J_{0.2} = 436$  N/mm for L360NE steel. For making the results more illustrative we shall determine critical depth  $a_{cr}$  of a long axial crack, e.g.  $2c = 120$  mm, for a pressure of 6.3 MPa, namely for the seamless pipe and for the

welded pipe. By fracture analysis it can be obtained  $a_{cr} \sim 4.2$  mm for the seamless pipe and  $a_{cr} \sim 3.6$  mm for the welded pipe. Although the critical crack depth  $a_{cr}$  for the seamless pipe is bigger its relative depth  $a/t = 0.494$  is smaller than that of the welded pipe ( $a/t = 0.571$ ). At the same time the pipe from L360NE steel is handicapped by its wall thickness which is about one third smaller than that of the pipe from 13126 steel. There is a problem of the life of a pipeline during which a crack grows to the critical depth. In order to quantify this process it will be supposed that the crack grows by a fatigue mechanism and the number of stress cycles required for the crack to reach the critical depth  $a_{cr}$  will be determined. Let us assume an initial axial crack of the length  $2c = 120$  mm and depth  $a = 2$  mm in each pipe. For application of the fatigue mechanism it will be assumed a fluctuation of the internal pressure in the range 1 – 6.3 MPa. Using experimentally determined Paris law constants it can be arrived at  $N_f = 4\,307$  cycles for the seamless pipe and  $N_f = 1\,651$  cycles for the welded pipe. This shows that the life of the seamless pipe would be about 2.6 times longer than that of the welded pipe.

If however, the thickness of the wall was the same for both pipes, e.g.  $t = 6.3$  mm, the critical crack depth in the seamless pipe would be only  $a_{cr} = 2.25$  mm and the number of cycles needed for a crack of the initial crack depth  $a = 2$  mm to reach critical depth 2.25 mm would be only 320 cycles.

### 4. Conclusions

These results show that even older pipelines constructed from pipes made by the seamless technology can compete with modern pipes manufactured from rolled out sheets. It is so when lowered strength and fracture properties are outweighed by a thicker wall.

### Acknowledgements

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# EXPERIMENTAL INVESTIGATION OF RESTITUTION COEFFICIENT FOR HYBRID COMPOSITES

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## 1. Introduction

Nowadays technology allows production of hybrid structures based on carbon, glass, aramid and wood-like materials. An important problem is to determine their mutual relations and mechanical properties of the final product. It is not possible to talk about the additive properties of the materials used here, because the composite made of them takes on material features only after the laminate has been shaped.

These types of laminates are widely used. Therefore, apart from the traditionally determined parameters for construction materials, it is also important to determine other parameters related to, for example, energy dissipation and vibration damping processes. Therefore, it is important to determine the restitution ratio for the newly created material. The restitution ratio is one of the physical features that characterizes the material's response to impact forces.

The aim of the work was to experimentally determine the coefficient of restitution (COR) for plates made of different hybrid composite materials.

## 2. Method and methodology

The scheme of the experiment set-up is shown in Fig. 1. It consists of: an impactor fitted with an acceleration sensor; a guide pipe for the impactor; a stand to fix the pipe; a steel plate with a square cutout on which the samples were fixed.

The impactor used in the tests had the form of a steel cylinder with a hemisphere ending. The outer diameter of the spherical part of the impactor was 20 mm. On the other end of the impactor, a MEMS acceleration sensor was mounted. The weight of the hammer was 149.1 g.

The impactor was placed in a polycarbonate tube, which acted as a guide during the drops. The inner diameter of the pipe was 26 mm.

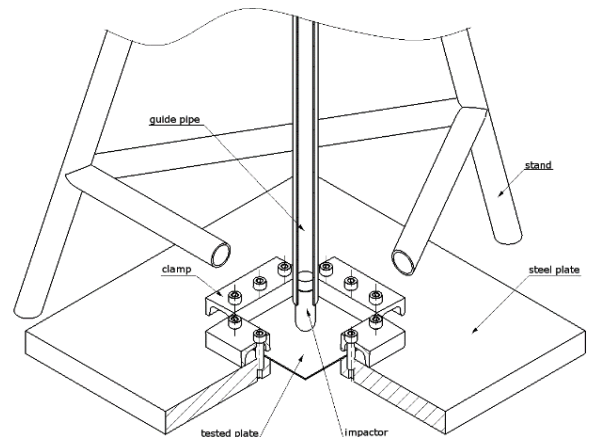


Fig. 1. The experiment set-up

Before the tests, the samples were attached to the plate shown in Fig. 2. In the central part of the plate, a square-shaped cut-out with a side length of 100 mm was made.

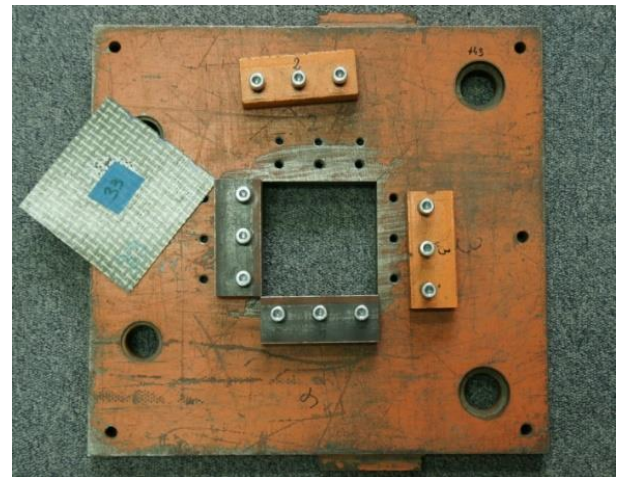


Fig. 2. The plate to which the tested plates were fixed

A line gauge was attached to the guide, which was used to determine the impactor's height after rebound.

Tests were carried out for 10 kinds of plates, made of five types of composites manufactured in two technologies: contact lamination and vacuum lamination.

The impactor was dropped in successive attempts from the height of 100, 200, 300 mm with video recording and from the height of 350, 400, 450 and 500 mm without video recording.

Measured coefficient of restitution:

$$\varepsilon_{meas} = \sqrt{\frac{h'_{meas}}{h_{meas}}} \quad (1)$$

$h'_{meas}$  – height achieved by the hammer after rebound;  $h_{meas}$  – height of the impactor drop.

Assuming that the energy of the drop hammer during the test is only dissipated when it hits the plate, it is possible to determine the height of the hammer based on the time between the individual impacts. The following relationship was used:

$$h_{cal} = \frac{1}{8} g \tau^2 \quad (2)$$

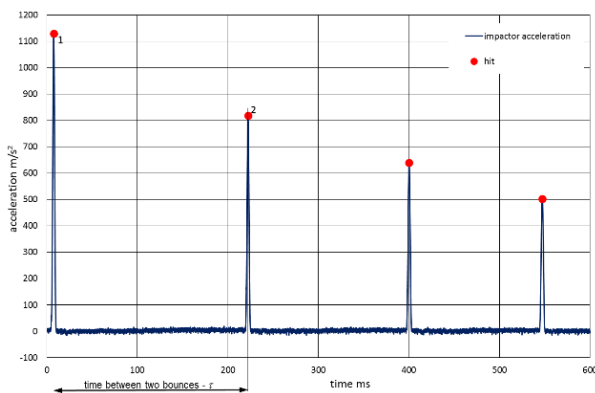
$g$  – earth gravity;  $\tau$  – time between rebounds.

Computational coefficient of restitution:

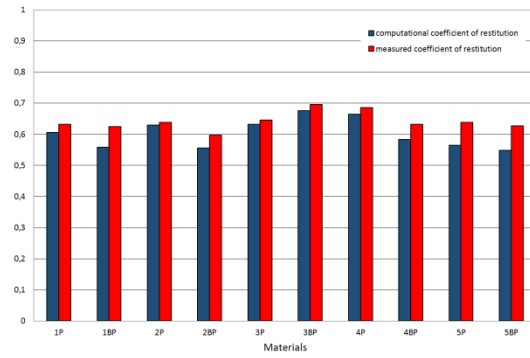
$$\varepsilon_{cal} = \sqrt{\frac{h'_{cal}}{h_{cal}}} \quad (3)$$

### 3. Results

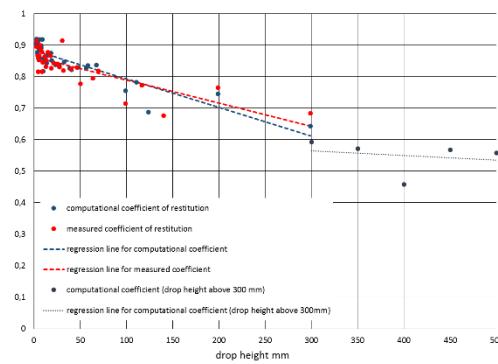
The accelerations acting on the impactor, recorded during the tests, were used to determine the coefficient of restitution of the examined plates (Fig. 3). Each hit of the plate is clearly visible in the recorded impactor acceleration. The following peaks observed in the graphs correspond to the next hits. On this basis, it was possible to determine the time between successive rebounds of the impactor.



**Fig. 3.** Recorded acceleration of the impactor. Drop from a height of 100 mm; the time between the first and the second hit is shown



**Fig. 4.** Comparison of the COR values for the calculation and measurement coefficients (drop height 300 mm)



**Fig. 5.** The COR values with regression lines

### 4. Conclusions

Two algorithms for the determination of the coefficient of restitution were used: rebound height measurement and calculation based on the time between successive bounces. Analysis of the COR results reveals that its value depends on the height from which the impactor was dropped. It was confirmed that this relationship is close to linear with a small negative directional coefficient of simple regression. It can be noticed that in the case of a computational coefficient, the slope angle of the regression line is greater than in the case of the measurement coefficient. As a result, for larger drop heights (above 300 mm) the calculation COR is lower than the measurement COR, and for smaller heights (below 200 mm) the situation reverses.

Both methods used to determine the coefficient of restitution gave similar results.

### Acknowledgement

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# DAMAGE MONITORING OF DIFFERENT CONCENTRATION CARBON NANOTUBE/EPOXY GLASS FIBER REINFORCED COMPOSITES UNDER QUASI-STATIC INCREMENTAL LOADINGS

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## 1. Introduction

The continuous demand for advanced materials, coupled with resources-saving requirements, has led aerospace industries to introduce innovative approaches, in terms of mechanical performance and Structural Health Monitoring (SHM) of aircraft structures. Integrated solutions, meeting both requirements (i.e., increased performance and simultaneous SHM) are the subject of intensive scientific research, worldwide, for quite a long time. Several solutions have been proposed recently, addressing the problem, or coping, in general, with the multi-functionality concept. Baron and Schulte [1] reported that the electrical resistance change (ERC) of the electrically conductive carbon fiber reinforced polymers (CFRP) could be used to monitor their structural health. Nevertheless, on the absence of an electrically conductive medium, this concept cannot be used in non-conductive polymers, e.g. glass fiber reinforced polymers (GFRP).

The triggering point for the implementation of such concepts in non-conductive polymers and composites was the exploitation of carbon nanotubes (CNTs) during the last decade. CNTs are derivatives of carbon and therefore comprise excellent mechanical and electrical properties. The successful dispersion of CNTs in the polymeric matrix, could perhaps transform the non-conductive GFRPs to electrically conductive and therefore enhance their sensing capabilities. Several research articles have been published concluding that the ERC can detect the onset of damage [2,3]. On the other hand the fracture properties of the respective materials appear significantly enhanced [4,5].

Research on these multi-functional materials is mainly focused on the enhancement of their mechanical performance as well as on measurements of ERCs, simultaneously recorded with stress/strain

variances of the specimens tested [5]. Nevertheless, critical parameters such as the mechanical/electrical response under incremental cyclic loadings are scarcely reported in the literature and will be investigated in the present study.

## 2. Materials and Methodology

Multifunctional composites were manufactured with unidirectional structural S2 glass being the strengthening phase of the composite. The resin system used was a typical epoxy, Araldite LY 564. It was selected primarily due to its low viscosity and its high mechanical properties. For each different case, resin of approximate 400 g and different percentages of CNTs (multi-wall carbon nanotubes provided by Arkema) were mixed in the dissolver. A shear mixer was used to efficiently disperse the CNTs in the dissolver solution.

The dissolver's mixing time of the resin with the different percentages of CNTs, namely 0.5%, 0.75% and 3.0%, lasted more than 24 h each. The resin was then mixed with the catalyst (Aradur 2954 by Huntsman) and it was infused to the metallic mould to produce composite rectangular plates of dimensions 300 mm x 300 mm.

The plates were properly machined to provide rectangular specimens of width equal to 25 mm. Electrical cable connectors were attached on each specimen, corresponding to three different gauge lengths, equal to 25, 75 and 125 mm (Fig.1).

The specimens were then subjected to successive tensile loading-unloading loops with the aid of an INSTRON servo-hydraulic loading frame of capacity equal to 100 kN. The maximum stress level attained in each loop was increasing by a step of 50 MPa with respect to that of the previous loop.

The quantities continuously measured and recorded during the experiments included: The load

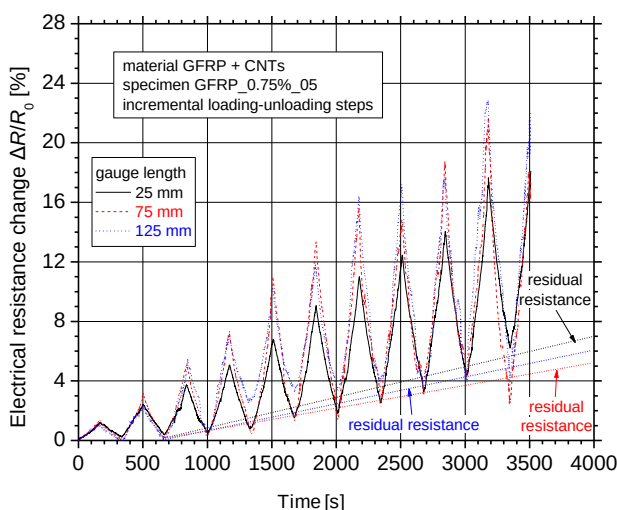
imposed (as it was provided by the calibrated load cell of the frame), the axial strain developed along the longitudinal axis of the specimens (provided by a calibrated INSTRON extensometer), and the Electrical Resistance Changes (ERC) for all three measuring lengths.



**Fig. 1.** A typical specimen, gripped in the loading frame, while being tested. The measuring system for all three gauge lengths is clearly seen, together with the extensometer used to measure the axial strains developed.

### 3. Results and Discussion

The time evolution of the Electrical Resistance Change (ERC) is plotted in Fig.2 for a typical specimen with 0.75 wt % CNTs.



**Fig. 2.** The electrical resistance change versus time for the three gauge lengths.

It is observed from Fig.2 that the maximum value of the ERC recorded for each loading loop increases according to an almost linear manner. Moreover, a residual ERC is detected for all three gauge lengths. This residual ERC increases gradually, also, according to an almost linear manner, systematically depending on the gauge length. Slight discrepancies observed are well within experimental error.

### 4. Conclusions

The residual ERC values after each unloading branch provide evidence that the selected surface sensor mimics the response of the residual axial strain values of the specimen. Correlation of the residual ERC change, after each unloading branch, with the respective residual axial strain values of the composite is analytically discussed in the article.

### Acknowledgements

The authors kindly acknowledge the contribution of Dr. Z. Metaxa in the preparation of the specimens and the experimental protocol

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# DESIGN OF COMMERCIAL HEXAGONAL HONEYCOMBS OF EQUAL OUT-OF-PLANE SHEAR MODULI

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## 1. Introduction

Honeycomb cores used in sandwich plates have an important contribution to the plate deflections that are emphasized through the homogenized out-of-plane shear moduli  $G_{13}$  and  $G_{23}$ . The honeycombs with uniform cell walls thickness have  $G_{13} = G_{23}$ , but honeycombs with double vertical walls thickness (commercial) do not exhibit this property, [1]. Recently, [2], the authors of this paper showed that it is possible to design in-plane “isotropic” hexagonal honeycombs. In this paper we analyze the conditions for which a general commercial honeycomb may become out-of-plane “isotropic” for shear moduli only, i.e.  $G_{13} = G_{23}$ .

## 2. Analytical approach

Homogenized elastic properties of honeycombs (see Fig. 1) are usually obtained using the beam theory [2, 3] for  $t \ll \ell$ .

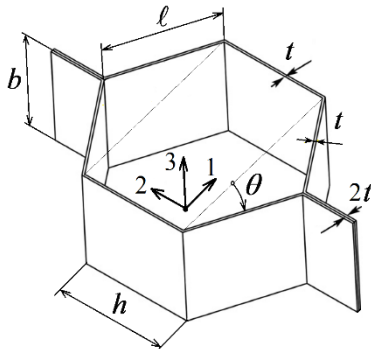


Fig. 1. Geometry of honeycomb cell.

The relative density of the equivalent core is

$$\frac{\rho}{\rho_s} = \frac{\frac{t}{\ell} \left( \frac{h}{\ell} + 1 \right)}{\left( \frac{h}{\ell} + \sin \theta \right) \cos \theta}, \quad (1)$$

where  $\rho$  and  $\rho_s$  are the mass densities of honeycomb core and of solid cell wall material. The two out of

plane shear moduli can be obtained from the following equations:

$$G_{13} = G_s \left( \frac{t}{\ell} \right) \frac{\cos \theta}{\left( \frac{h}{\ell} + \sin \theta \right)}; \quad (2)$$

$$G_{23} = G_{23}^{LB} + \frac{\alpha}{\left( \frac{b}{\ell} \right)} \left( G_{23}^{UB} - G_{23}^{LB} \right), \quad (3)$$

where

$$G_{23}^{UB} = G_s \left( \frac{t}{\ell} \right) \frac{\frac{h}{\ell} + \sin^2 \theta}{\left( \frac{h}{\ell} + \sin \theta \right) \cos \theta}; \quad (4)$$

$$G_{23}^{LB} = G_s \left( \frac{t}{\ell} \right) \frac{\frac{h}{\ell} + \sin \theta}{\left( \frac{h}{\ell} + 1 \right) \cos \theta}, \quad (5)$$

and according to [4] and [5]

$$\alpha = \begin{cases} 0.787 & \text{for } \theta \geq 0^\circ \\ 1.342 & \text{for } \theta < 0^\circ \end{cases}. \quad (6)$$

From (2) and (3), imposing  $G_{13} = G_{23}$ , and using the notations  $x = h/\ell$ ;  $c = \cos \theta$ ;  $s = \sin \theta$  and  $k = \alpha \ell/b$  it yields the equation:

$$\frac{c}{x+s} = \frac{x+s}{c(x+1)} (1-k) + k \frac{x+s^2}{c(x+s)}, \quad (7)$$

with only two unknowns  $x$  and  $\theta$ , if we suppose that  $k$  is specified. The solutions of (7) must fulfill the geometrical conditions

$$x = \frac{h}{\ell} \geq 0 \quad \text{and} \quad \begin{cases} -\sin^{-1} \frac{x}{2} < \theta < 90^\circ & \text{for } x \leq 2 \\ -90^\circ < \theta < 90^\circ & \text{for } x > 2 \end{cases}. \quad (8)$$

Essential unknowns are typically  $x$  and  $\theta$ , and any of these may initially be chosen for a design configuration as needed. The  $k$ -parameter depends

on both the angle  $\theta$  and the ratio  $\ell/b$ , which initially is not known, suggesting that iterative calculations are sometimes required for a complete solution.

If  $k$  and  $\theta$  are known, (7) becomes a second-degree algebraic equation in  $x$  for which possible solutions become

$$x_{1,2} = s(k-1) - \frac{k}{2}(1+s^2) + \frac{c^2}{2} \pm \frac{\sqrt{\Delta}}{2}, \quad (9)$$

if

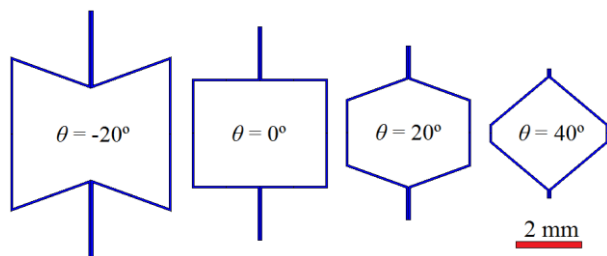
$$\begin{aligned} \Delta = & c^2(c^2 - 4s + 4) + \\ & 2k(2s^3 - c^2s^2 + 2c^2s - c^2 - 4s^2 + 2s) + \\ & k^2(s^4 - 4s^3 + 6s^2 - 4s + 1) \geq 0. \end{aligned} \quad (10)$$

### 3. Analytical results

For a material set of given data  $E_s = 70$  GPa,  $\nu_s = 0.33$  and  $\rho_s = 2700$  kg/m<sup>3</sup>, an imposed cell wall thickness  $t = 0.0635$  mm, height of the honeycomb core  $b = 30$  mm, and constant effective mass density of the honeycomb core  $\rho = 135$  kg/m<sup>3</sup> (see Eq. (1)), using the presented relationships, some results are presented in Table 1 and Fig. 2.

**Table 1.** Some particular geometric configurations and out-of-plane shear moduli obtained for given data.

| $\theta$<br>[°] | $x = \frac{h}{\ell}$<br>[-] | Analytic                 | FEA               |                   |
|-----------------|-----------------------------|--------------------------|-------------------|-------------------|
|                 |                             | $G_{13}=G_{23}$<br>[MPa] | $G_{13}$<br>[MPa] | $G_{23}$<br>[MPa] |
| -20             | 1.786                       | 417.1                    | 417.6             | 393.1             |
| 0               | 1.579                       | 510.2                    | 510.9             | 512.6             |
| 20              | 0.968                       | 590.5                    | 591.3             | 590.3             |
| 40              | 0.193                       | 647.2                    | 647.9             | 646.6             |



**Fig. 2.** Four possible solutions for given data.

### 4. Validation using numerical approach

Analytically obtained results in the previous paragraph were verified with FEA using the Shell-finite element models presented in [6, 7]. For the four analytical solutions we can see that the numerical values for the shear moduli (columns 4 and 5 of Table 1) are very close to the analytical

solutions. Larger, but insignificant, errors are obtained for low values of  $\theta$ , because the relation (6) for  $\theta < 0$  is obtained in [5] by a least-square technique over 30 different calculated moduli, with large scattering. The coefficient  $\alpha = 1.342$  is not fully correct for particular values of considered given data, and for this reason, in the case of pronounced re-entrant honeycomb, no perfect equality  $G_{13} = G_{23}$  was achieved.

### 5. Conclusions

According to authors' knowledge, this paper presents for the first time the design of an out-of-plane isotropic commercial hexagonal honeycomb, both analytically and afterwards numerically, as to describe the complete geometry of a cell. It can be observed from the presented results that as cell angle  $\theta$  increases, the ratio  $x = h/\ell$  decreases but the shear modulus increases.

### Acknowledgements

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# A SMALL-SCALE TEST BRIDGE FOR MEASUREMENT- AND MODEL-BASED STRUCTURAL ANALYSIS

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## 1. Introduction

To examine the capability to detect and localise damage using the Measurement- and Model-based Structural Analysis (MeMoS) as presented in [1], a small-scale truss bridge (1520 mm × 720 mm × 720 mm) made of aluminium profiles is built as a test specimen for this purpose. The truss frame of the test bridge is made of aluminium profiles with a sophisticated design of the cross-sectional area. In comparison, with solid profiles, only a fraction of the material is needed to produce the profiles, while their bending resistance decreases slightly. The profiles are built into a truss frame by connecting them by means of fastening sets made of steel. The bridge model is mounted on four steel bearings which each of them consist of a cylinder arranged between two plates. Fixed bearings are made by holding onto one end of the bridge. The bridge is subjected by an external load by placing a heavy object beneath it. At the same time, measurements can be conducted below the bridge. Therefore, the bridge specimen is elevated by attaching it on a pedestal with four columns. Damages can be induced by loosening the fastening pieces. The set-up is shown in Fig. 1.



Fig. 1. The bridge specimen on the pedestal, approximately 2147.6 N was applied

## 2. Measurement equipment

*Photogrammetry* is applied to take measurements from photographs. The absolute positions of surface markers on an object of interest in each photo are determined. The evaluation of photos taken from two different states results the displacement of markers, i.e. the position change of a specific marker between the reference and current states. *Load cell* is used to measure force. When a force is applied on a load cell, certain structures within the cell deform. And this deformation is captured by strain gauges. The force can in turn be determined by the calibrated strain gauges.

## 3. Measurement of displacements

Round target stickers applied on surface of the bridge specimen were used to track local displacements. By means of a commercial photogrammetry system (GOM Correlate), many photos of the bridge specimen can be taken from different viewing points and angles. And in a post-processing procedure, the coordinates of the markers were determined. The positions of the marker points were determined for different load and damage states of the bridge model. The experiment was conducted for five different states: (1) Undamaged and unloaded, (2) undamaged and loaded, (3) damage level 1 and loaded, (4) damage level 2 and loaded, (5) damage level 3 and loaded. In the first damage level three screws are loosened. In the second stage, three additional screws are released in addition to the first one. In damage level 3, in total ten screws are removed. The displacements of the bridge specimen are determined with respect to the undamaged and unloaded state (1).

## 4. Calibration of the reference state

The approximate substitute model from [2] is used to describe the aluminium profiles of the

bridge specimen since a finite element discretisation of the complete body would be infeasible. These aluminium profiles are longitudinal objects with a very detailed cross section. This requires a finite element discretisation with impracticable element quantity. This in turn requires large amounts of computer memory that is not available. The connection parts are not further examined and are therefore ignored. Consequently, the bridge specimen is fully characterised by six material constants. For the reference state of the bridge model, the six unknown material parameters and their stochastic properties are determined from the displacements from state (1) to (2).

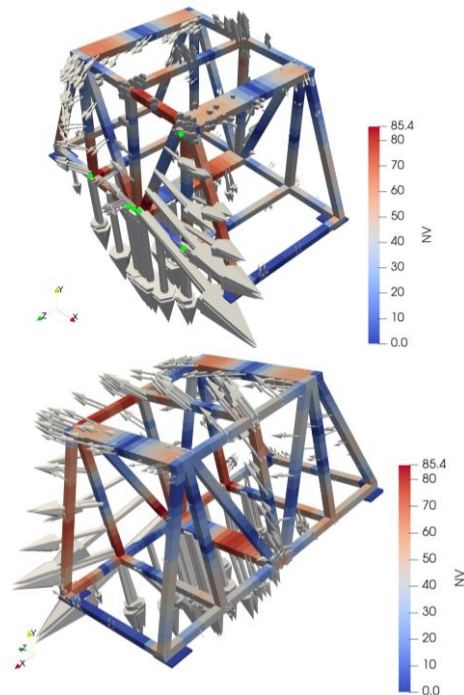
## 5. Damage detection and localisation

To detect and localise damage, the approach presented in [1] is followed. But, some modifications must be made due to practical considerations. The bridge specimen's approximate finite element model consists of 545495 elements. To reduce the computational time, an alternative finite element model is grouped into 598 chunks. For each of these chunks, a separate set of unknown material parameters is introduced into the adjustment as observed unknowns.  $\mathbf{L}_i$  is the vector of displacement observations for the set  $i$ . The different sets of displacement measurements are:  $\mathbf{L}_1$  from state (1) to (2),  $\mathbf{L}_2$  from state (1) to (3),  $\mathbf{L}_3$  from state (1) to (4) and  $\mathbf{L}_4$  from state (1) to (5). After the adjustment it is possible to perform a global test respectively  $\chi^2$ -test to detect damage. As expected if  $\mathbf{L}_1$  is reused to adjust the elastic parameters and subsequently performing the global test, we fail to reject the null hypothesis that no damage has occurred. Furthermore, when the other displacement measurements  $\mathbf{L}_2$ ,  $\mathbf{L}_3$  and  $\mathbf{L}_4$  are evaluated, we reject the null hypothesis in favour of the alternative hypothesis. This indicates that damages are detected for the corresponding cases. To localise the damage, it is proposed here to evaluate the *standardised residuals* NV of the observed unknowns for the cases where the null hypothesis is rejected in the global test. The largest standardised residual will lead to the location of the damage.

## 6. Results

The standardised residuals NV of the observed unknowns for the displacement measurements  $\mathbf{L}_4$  are shown in Fig. 2. The displacement measurements  $\mathbf{L}_4$  capture the bridge specimen, where all ten screws are removed. Since on one side

(Fig. 2 top) more screws are loosened than on the other side (Fig. 2 bottom), it is to be expected that one side with most of the removed screws has the largest displacement field. This in turn yields large standardised residuals. It can be observed on this side in Fig. 2 (top) of the specimen that a distribution of large standardised residuals can be found.



**Fig. 2.** The standardised residuals NV of the observed unknowns by evaluation of the displacement measurements including the observed displacement vectors magnified 500 times, green dots indicate the damages

The global test is very well suited to determine whether there is any damage at all. The localization of individual damages with the help of the local test is in general possible, but in practice it is made difficult by the influence of systematic errors (e.g. non-modelled properties of the mechanical model). However, MeMoS can indicate the area of damage, even if it is not able to pinpoint the exact location of the damage in the examinations carried out.

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## FATIGUE DAMAGE PREDICTION OF NOTCHED SAMPLES

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### 1. Introduction

The design of the mechanical components inevitably involves geometries with discontinuities of various shapes, generally referred to as *notches*. These, in turn shows distortions in the stress waves flux highlighted by the high stress gradients and often called as *the stress concentration effects*.

The notch effect is quantified by the theoretical stress concentration factor,  $K_t$ , in case of the static loadings in the elastic domain of the material. For fatigue loadings, the notch effect is cumulated with the mechanical behavior of the material in the so-called notch sensitivity and is quantified by the fatigue notch factor,  $K_f$ . The problem of the notch effect is a key issue in predicting the durability of the mechanical components.

For materials with elastic-plastic behavior, Neuber, [1], has proposed a rule that allows the estimation of the real stress and strain state at the notch tip. This rule has proven to be very effective in analyzing the mechanical behavior of the notched components. It has also been used in various applications, and even implemented in FEM analysis softwares, (e.g. Ansys).

In this paper, the Neuber's rule was used to predict the durability of notched samples. Further, the durability estimation was made based on the strain life approach using a Coffin-Manson-Basquin curve determined based on the mechanical properties of the material.

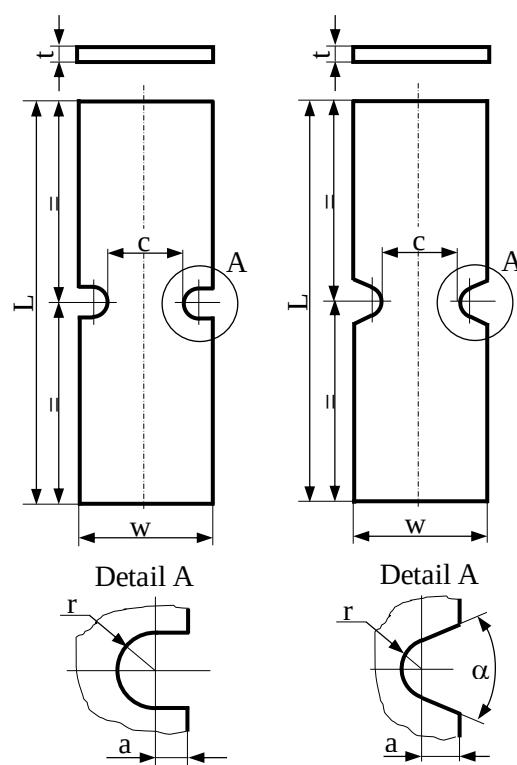
### 2. Material and methods

The material analyzed in this paper is a low carbon steel, for which tensile tests and Brinell hardness measurements have been performed. The mechanical properties of the material are given in Table 1. Also, fatigue tests on notched samples have been carried out on a fatigue testing machine Walter-Bai. The fatigue tests were performed on sinusoidal cycles with stress ratio  $R = 0.1$ . Two notch shapes, U and V, were analyzed on samples

with different widths. The sample sizes are given in Figure 1 and Table 2. For each notch shape, the theoretical stress concentration factor and respectively the fatigue notch factor have been determined and given in Table 3.

**Table 1.** Mechanical properties of the low carbon steel

| Young's modulus [MPa] | Yield strength [MPa] | Ultimate strength [MPa] | Brinell hardness [HB] |
|-----------------------|----------------------|-------------------------|-----------------------|
| 191000                | 160                  | 281                     | 80                    |



**Fig. 1.** The notched samples used in the fatigue tests

**Table 2.** The sizes of the notched samples

| Notch shape | Dimensions [mm] |    |   |     |    |     | $\alpha$ [°] |
|-------------|-----------------|----|---|-----|----|-----|--------------|
|             | L               | w  | t | a   | c  | r   |              |
| U           | 150             | 25 | 3 | 2,5 | 15 | 2,5 | -            |
|             |                 | 30 |   |     | 20 | 2,5 | -            |
| V           | 150             | 25 | 3 | 2,5 | 15 | 2,5 | 60           |
|             |                 | 30 |   |     | 20 | 2,5 | 60           |

**Table 3.** The stress concentration factors

| Notch shape | Width [mm] | Theoretical stress concentration factor $K_t$ | Fatigue notch factor $K_f$ |        |
|-------------|------------|-----------------------------------------------|----------------------------|--------|
|             |            |                                               | Peterson                   | Neuber |
| U           | 25         | 2,38                                          | 2,25                       | 1,92   |
|             | 30         | 2,63                                          | 2,48                       | 2,09   |
| V           | 25         | 2,22                                          | 2,11                       | 1,82   |
|             | 30         | 2,44                                          | 2,31                       | 1,96   |

According to Neuber's rule it is assumed that the material at the notch tip exhibit an elastic-plastic behavior that can be described by the cyclic stress-strain curve. Considering this, the real stress and strain state at the notch tip is different than indicated in a linear elastic analysis and can be determined by the intersection of the cyclic stress-strain curve with the Neuber's hyperbola. Analytically, the intersection point can be determined by solving the following system of equations:

$$\begin{cases} \frac{(K_t \cdot S_{n,max})^2}{E} = \sigma_{max}^r \cdot \varepsilon_{max}^r \\ \varepsilon_{max}^r = \frac{\sigma_{max}^r}{E} + \left( \frac{\sigma_{max}^r}{K'} \right)^{\frac{1}{n'}} \end{cases} \quad (1)$$

where:

$K_t$  is the theoretical stress concentration factor,  $S_{n,max}$  is the nominal maximum stress,  $E$  is the Young's modulus,  $\sigma_{max}^r$  and  $\varepsilon_{max}^r$  are the maximum real tensile stress and strain at the notch tip and  $K'$  respectively  $n'$  are the cyclic material parameters in the Ramberg-Osgood equation. The cyclic material parameters are estimated according to Reference [2].

### 3. Results and discussions

Fatigue life predictions of notched specimens were performed based on Smith-Watson-Topper (SWT) model and the strain life equation. According to this model the fatigue life is controlled by crack growth on planes perpendicular to the maximum tensile stress:

$$\sigma_{max} \cdot \frac{\Delta \varepsilon}{2} = \frac{(\sigma_f')^2}{E} \cdot (2N)^{2b} + \sigma_f' \cdot \varepsilon_f' \cdot (2N)^{b+c} \quad (2)$$

where,  $\sigma_{max}$  and  $\Delta \varepsilon$  represents the real maximum stress and strain range at the notch tip determined by the Neuber's rule and  $\sigma_f'$ ,  $\varepsilon_f'$ ,  $b$  and  $c$  are material fatigue parameters. Also, the material fatigue parameters were estimated according to the correlation models given in Reference [2].

Table 4 shows the results of fatigue life prediction analysis for the two types of notched samples together with the corresponding experimental results.

**Table 4.** Fatigue life predictions of notched specimens

| Method     | Parameter   | Notch  |        |       |        |
|------------|-------------|--------|--------|-------|--------|
|            |             | U      |        | V     |        |
|            |             | w30    | w25    | w30   | w25    |
|            | $S_{n,max}$ | 269.36 | 271.18 | 220.3 | 202.02 |
| M-M        | $N_{pred.}$ | 6584   | 9488   | 20133 | 45400  |
| B-S        | $N_{pred.}$ | 4552   | 6314   | 12262 | 24705  |
| R-F        | $N_{pred.}$ | 17607  | 25293  | 53260 | 118861 |
| $N_{exp.}$ |             | 4547   | 13504  | 74925 | 73462  |

M-M – Muralidharan-Manson; B-S – Baumele-Seeger; R-F – Roessle-Fatemi, [2].

The results indicate reasonable predictions of the fatigue life of notched samples, considering that the analysis was made without knowing any fatigue data of the material. The predicted fatigue lives are within factor of 2 scatter bands for M-M and R-F methods and within factor of 3 scatter bands for B-S method. For each method there was an exception in which the predicted fatigue life was out of factor of 3 scatter bands. Thus, for the M-M and B-S methods, the exception was for the V-notch sample of 30 mm width and for the R-F method the exception was at the U-notch sample of 30 mm width.

### 4. Conclusions

In this paper an analysis of fatigue life prediction of notched samples is presented, in the absence of any fatigue data of the material. Reasonable predictions were obtained compared to experimental results. At the same applied nominal stress, a greater severity of the U-notch was observed over the fatigue life, compared to the V-notch. This is due to the stress concentration factor which is higher at the U-notch.

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# STIFFNESS DETERMINATION IN STEEL SPECIMENS ORIGINATING FROM NEW AND DISCARDED TRAMWAY RAILS BY MEANS OF NANOINDENTATION

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## 1. Introduction

In urban areas, tramways are an important means of transportation demanding high reliability and safety of the whole tramway network. The tramway rails, which typically exhibit a groove as they are embedded into the pavement, are made of self-hardened steel. Failure of these rails is clearly caused by insufficient mechanical properties, due to which the loads, the rails are subjected to during operation, cannot be carried as intended. In this context, it is important to mention that several factors may lead to deterioration of the rails during their service life, facilitating failure. E.g. in the course of the manufacturing process, eigenstresses are induced, which, during tram operation, superpose with the mechanical stresses arising from the axle loads of tramways, as well as with stresses due to temperature changes. In addition, tramway rails experience material degradation due to wear and environmental influences such as corrosion.

This contribution focusses on one major mechanical property, namely the stiffness. By means of a range of experimental techniques, comprising nanoindentation and light microscopy, the stiffness of new and discarded tramway rails is studied.

## 2. Materials and Methodology

### 2.1 Investigated tramway steel

In the Viennese tramway network, from fractured rails that cannot be repaired by means of welding, about 700 mm long rail pieces including the fracture are removed and replaced by new pieces of rail. In this study, we investigate old discarded rails exhibiting fractures in the welding joints. Therefore, three of these discarded rails of the steel grade S800 and the grooved profile 60R3 were

collected, installed in the years 1973, 1979 and 2006. Two approximately 1 cm thick sections were cut from these pieces, one located in the heat-affected zone (HAZ) next to the welding joint, and one located outside of the HAZ. In addition, one section of a new rail was taken as reference.

### 2.2 Sample preparation

To obtain the distribution of the Young's modulus over more or less the whole cross section, six specimens were cut out from each section, see for instance Fig. 1. In order to obtain a flat and smooth surface finish for nanoindentation, all specimens were ground and polished in eleven stages of increasing fineness, starting with a grain size of 196  $\mu\text{m}$  down to a grain size of 1  $\mu\text{m}$ , according to [1]. Between the stages, the specimens were cleaned by ultrasonics to avoid contamination by coarser abrasive.

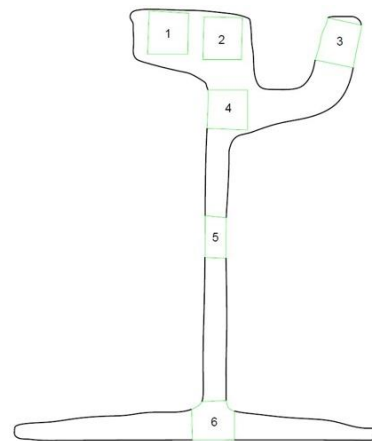


Fig. 1. Location of the six specimens in rail 60R3.

### 2.3 Nanoindentation testing

The nanoindentation tests were performed on the polished samples, using a TriboIndenter nanoindenting system (Hysitron Inc. Minneapolis, MN, USA) equipped with a Berkovich diamond tip.

The mesh grid of each specimen comprised 20 x 20 indents with a grid spacing of 0.5 mm. A trapezoidal load history was prescribed, defined by a loading time of 30 s, a holding time of 3 s and an unloading time of 30 s, applying a maximal indentation load of 6 mN. The reduced Young's modulus  $E_r$  was obtained according to Oliver and Pharr [2],

$$E_r = \frac{S\sqrt{\pi}}{2\sqrt{A}} \quad (1)$$

where  $S$  describes the initial slope of the unloading path of the load-displacement curve, and  $A$  represents the projected area of the elastic indentation contact. With the Poisson's ratio and the Young's modulus of the Berkovich tip,  $\nu_t = 0.07$  and  $E_t = 1141$  GPa, the Young's modulus of the steel specimen can be determined when considering the Poisson's ratio for steel,  $\nu_s = 0.3$  [3],

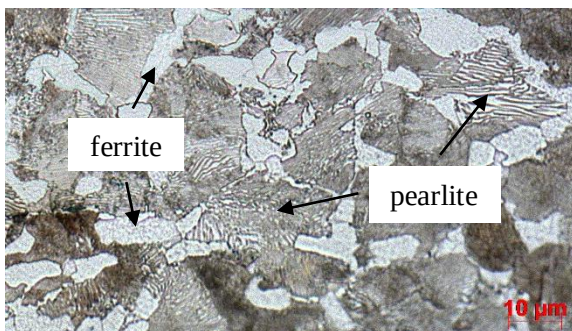
$$E_s = \frac{1-\nu_s^2}{\frac{1}{E_r} - \frac{1-\nu_t^2}{E_t}} \quad (2)$$

#### 2.4 Light microscopy

In addition, the microstructure was studied using a Zeiss Axio Imager capturing the images with a digital camera (the AxioCam MRc5). Prior to the light microscopy, the samples were etched for about 20 s in 2 % nital in order to visualize the microstructure of tramway rails.

### 3. Results

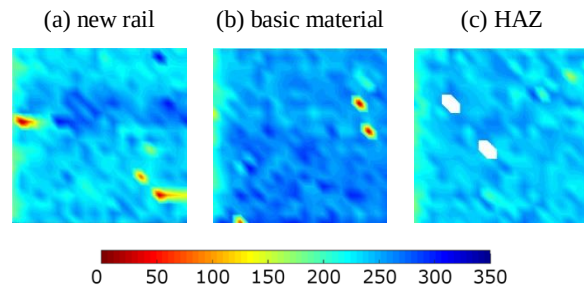
The pearlitic-ferritic microstructure of a new tramway rail with the steel grade S800 is shown in Fig. 2. The pearlite consists of ferrite and cementite lamellae.



**Fig. 2** Microstructure of new rail with steel grade S800.

The distributions of the Young's modulus for three rail specimens of the steel grade S800, which are located in position 4, see Fig. 1., are illustrated in Fig. 3. While Fig. 3(a) shows the results for a new rail, Fig. 3(b) and 3(c) depict the results for a 45 years old rail, installed in 1973. The specimen illustrated in Fig. 3(b) originates from the basic

material, while the specimen illustrated in Fig. 3(c) stems from the HAZ next to the welding joint. The red spots signify values where either the indentation tip or the specimen might have been contaminated, and the white spots signify values which are out of scale.



**Fig. 3.** Distributions of the Young's modulus in specimens located in position 4: (a) new rail; (b), (c) 45 years old tramway rail.

Except for these outliers, the investigated steel specimens exhibit mostly homogeneous stiffness distributions.

### 4. Conclusions

Building on nanoindentation tests and light microscopy, the key novelty of this paper is the investigation of new and discarded tramway rails, with a special focus on the heat affected zone and basic material. As mentioned above, the resulting distributions of the Young's modulus each show a homogenous distribution, allowing to conclude that maintenance measures to which a rail is subjected during its service life does not significantly affect the cross-sectional distribution of stiffness. Hence, rail failures are unlikely to be caused by stiffness irregularities.

### Acknowledgements

The financial support by Wiener Linien GmbH & Co KG, as well as their providing of testing material is gratefully acknowledged.

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# NUMERICAL AND EXPERIMENTAL INVESTIGATION OF CONTACT LENGTH DURING ORTHOGONAL CUTTING

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## 1. Introduction

In this contribution, we investigate the variation of tool–chip contact length at different chip thicknesses during orthogonal cutting. The contact length is the distance over which the chip remains in contact with the rake face of the tool. This is an important parameter in metal cutting, since it indicates the heat transfer between the tool and the chip, which dominates tool wear. Additionally, since the cutting force is the resultant of a force system distributed along the rake face, the contact length plays an important role in cutting force models and affects the occurrence of machine tool vibrations. Consequently, estimation and modelling of contact length is important in terms of understanding the dynamics of cutting [1-3].

## 2. Contact length models

In the literature, several contact length models are available, which are usually functions of the uncut chip thickness  $h$ . Some models (Altintas [1], Lee and Shaffer, Vinogradov [3]) are also dependent on shear angle  $\phi_c$  and tool geometry as

$$L_c^{\text{Alt}} = \frac{h \sin(\phi_c + \beta_a - \alpha_r)}{\sin \phi_c \cos \beta_a}, \quad (1)$$

$$L_c^{\text{Lee}} = \frac{2h \sin(\pi/4)}{\sin \phi_c \sin(\pi/4 + \phi_c - \alpha_r)}, \quad (2)$$

$$L_c^{\text{Vinogr}} = \frac{h \sin(\pi/4)}{\sin \phi_c \sin(\pi/4 + \phi_c - \alpha_r)}, \quad (3)$$

where  $\alpha_r$  is the rake angle of the tool and  $\beta_a$  is the average friction angle. The shear angle can be modelled based on the minimum energy principle (MEP) and the maximum shear stress principle (MSSP), which also depends on the tool geometry [1,4]. Parameters  $\phi_c$ ,  $\beta_a$ , and therefore the contact length  $L_c$  can be determined experimentally directly from measured cutting forces.

## 3. Experimental layout

The goal of the measurement was to determine experimentally the relationship between the uncut chip thickness  $h$  and the chip-tool contact length  $L_c$  using orthogonal planing tests [4]. The planing tests were performed on a single aluminum (A2024-T351) rib using the feed motion of the table of a NCT EmR-610Ms CNC milling machine. The cutting forces were measured using KISTLER 9129AA multicomponent dynamometer, while the whole chip formation process was recorded by Photron SA5 high-speed camera. The measurement layout and the applied parameters are presented in Fig. 1. and are listed in Table 1, respectively.

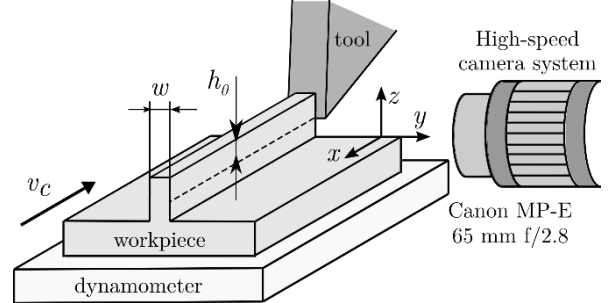


Fig. 1. The measurement layout

Table 1. Parameters of the measurement

|           |                                                                                                                 |
|-----------|-----------------------------------------------------------------------------------------------------------------|
| Tool      | Carbide tool,<br>$w_{\text{tool}} = 5 \text{ mm}$ ; $\alpha_r = 15^\circ$ ; $\alpha_f = 10^\circ$               |
| Workpiece | Aluminum A2024-T351,<br>rib thickness: $w = 2 \text{ mm}$                                                       |
| Process   | Orthogonal planing, $v_c = 10000 \text{ mm/min}$<br>$h_{\min} = 0.01 \text{ mm}$ ; $h_{\max} = 0.49 \text{ mm}$ |

The theoretical contact length expressions were evaluated with both shear angle models using the measured cutting forces. Additionally, the tool was covered with black painting, which the chip removed during cutting. This allowed the estimation of the maximal contact length via image analysis (see Fig. 2). This method was validated using high-speed camera recordings, which showed good agreement with the detected contact length values.

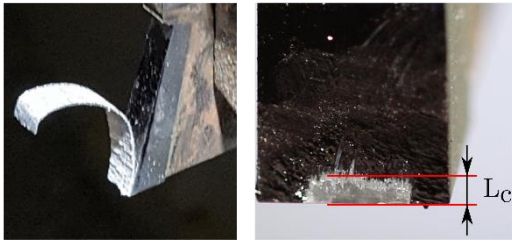


Fig. 2. The estimation of the contact length  $L_c$  using image analysis

The comparison of the contact length characteristics obtained from theoretical models and image analysis is presented in Fig. 3. The results shows that the model of Lee and Shaffer using the MSSP shear angle model gives the closest approximation to the detected contact length values.

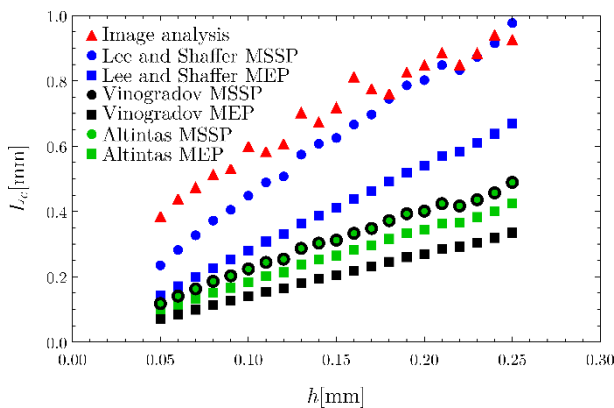


Fig. 3. Comparison of the average contact lengths based on theoretical models and the maximal contact length using image analysis at different chip thicknesses

#### 4. Validation using FE analysis

In order to validate the contact length models, we performed finite element simulations of the planing test. In the FE model (see Fig. 4.) Coupled Eulerian-Lagrangian (CEL) formulation was applied [5,6]. The mechanical behavior of the workpiece was modelled using the Johnson-Cook viscoplastic model in combination with the Johnson-Cook damage model, while Coulomb friction is adopted to model friction between the tool and workpiece.

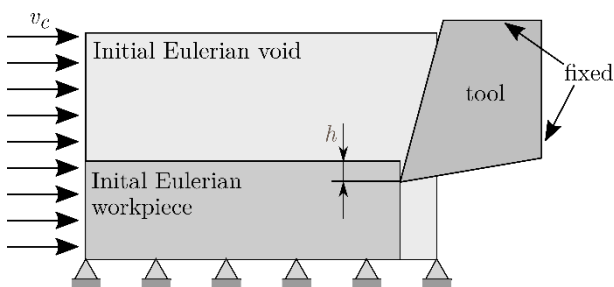


Fig. 4. The applied FE model using CEL formulation

#### 5. Conclusion

It can be concluded, that the contact length values as a function of the uncut chip thickness show linear characteristics. The experiments also revealed that the maximal contact length obtained from image analysis is in good agreement with the model of Lee and Shaffer combined with the MSSP shear angle model especially at high chip thickness values. At smaller values, however, the discrepancy between the models and the experiments becomes more significant. Additionally, the comparison of the measured and predicted contact length characteristics with the FE simulations using CEL formulation shows good agreement and helps to understand the dynamics of metal cutting.

#### Acknowledgements

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# ROOT ITERATION METHOD FOR MODAL PARAMETER IDENTIFICATION

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## 1. Introduction

This paper represents an intermediate state of an automatic modal identification tool that is able to extract modal behavior of a mechanical structure. These parameters are mandatory from vibrational monitoring and model developing perspective.

During the measurements one collects the so-called frequency response functions (FRF's) between various excitation and sensing points. All of these functions are collected into a 3D array which serves as input for the fitting algorithm. The algorithm uses the assumption of linearity for the curve fitting procedure. After the fitting was performed the modal parameters can be determined and stored for evaluation of the vibrational behavior or used for creating hybrid finite element models (FEM) based on the results of experimental modal analysis results.

The dynamic models based on the fitting algorithm are carrying the actual tested vibratory behaviour of the mechanical system. This results in a better simulation environment where the initial state of the system is ensured due to this characterization technique.

## 2. Fitting algorithm

The transfer function (TF) contains all information about the transient and stationary behavior of a given mechanical system. However, there is no possible way to actually measure TF.

### 2.1 FRF function in modal analysis

The FRF is a slice at  $s=i\omega$  of the TF. In connection with this relation we identify  $s$  and  $\omega$  being in Laplace and Fourier domain, respectively. Due to the multiple excitation and sensing points the TF can be given in the following, so-called right polynomial fraction form:

$$\mathbf{H}(s) = \mathbf{A}(s)\mathbf{B}^{-1}(s). \quad (1)$$

The modal parameters are hidden inside the poles, which are the singular points of the Transfer function. In the fractional form this means

$$\det \mathbf{B}(s) = 0. \quad (2)$$

The goal is to calculate the modal parameters such as the natural frequency  $\omega_n$  and damping ratio  $\zeta$  from  $s$

$$s_k = -\zeta_k \omega_{n,k} \pm i\omega_{n,k}\sqrt{1-\zeta_k^2} \rightarrow \omega_{n,k}, \zeta_k, \quad (3)$$

where  $k = 1, \dots, N$ . There is one thing only the FRF can be measured instead of a TF, but not even all necessary points can be reached during a real industrial test. Thus only a truncated version of the theoretical FRF  $\mathbf{H}(\omega)$  can be measured, which forms a non-symmetric matrix as

$$\Phi(\omega): R \rightarrow C^{m \times n}. \quad (4)$$

Eq. (4) describes the data input for the algorithm and (1) describes the fitting model which was used in the algorithm.

### 2.2 Mathematical model of the algorithm

According to (1), the FRF can be modeled as a fraction of two matrices as well, here: a non-symmetric  $\mathbf{A}$  and symmetric matrix functions  $\mathbf{B}$  [1]:

$$\mathbf{H}(\omega) = \mathbf{A}(\omega)\mathbf{B}^{-1}(\omega), \quad (5)$$

$$\mathbf{A}(\omega): R \rightarrow C^{m \times n}, \mathbf{B}(\omega): R \rightarrow C^{n \times n},$$

where each of the two matrices are made from Forsythe base polynomials  $P_l(\omega)$  [2], which are orthogonal (avoid numerical errors) and Hermitian symmetric (property of a real linear system). To do the fitting the following error term were defined and minimized ( $\mathbf{A}(\omega) := \sum_l \mathbf{A}_l P_l(\omega)$ ,  $\mathbf{B}(\omega) := \sum_l \mathbf{B}_l P_l(\omega)$ ), as

$$\mathbf{E}(\omega) = \mathbf{A}(\omega) - \Phi(\omega)\mathbf{B}(\omega) \text{ with} \quad (6)$$

$$J(\mathbf{A}_l, \mathbf{B}_l) = \text{tr } \mathbf{E}^H(\omega) \mathbf{E}(\omega) \rightarrow \min.$$

### 2.3 Calculation of the modal parameters

The results of (6) are the polynomial coefficients. Using that and the base polynomial terms, so-called comrade matrix [3] can be constructed, which eigenvalues are approximately the  $s_k$ 's, thus from ((3) the modal parameters ( $\omega_n$ ,  $\zeta$ ) are definable.

### 2.4 Measurement results

Putting all together,  $s_k$ 's calculated with several fitting polynomial order, from which the modal parameter, in this case the "founded" eigenfrequencies can be calculated. Plotting these eigenfrequencies for every fitting order the stability diagram can be constructed as shown in Fig. 2.

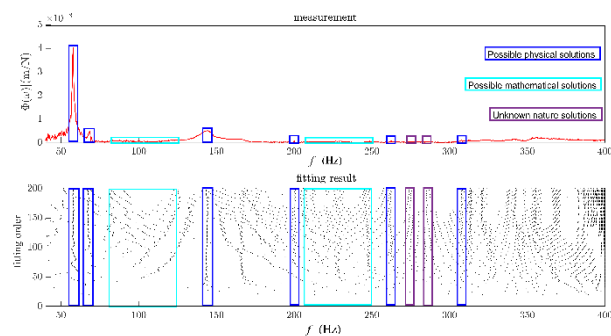


Fig. 2. Stability diagram.

From Fig. 2. it can be seen that for several cases the algorithm was able to identify the proper modal parameters: in this case the eigenfrequencies. But there were certain situation where it did wrong. This was because the fitting process is a root (poles) finding method after all. Some of the roots contain physical meaning i.e.: describes the structure's vibrational mode, others just arise with the mathematical nature of the method.

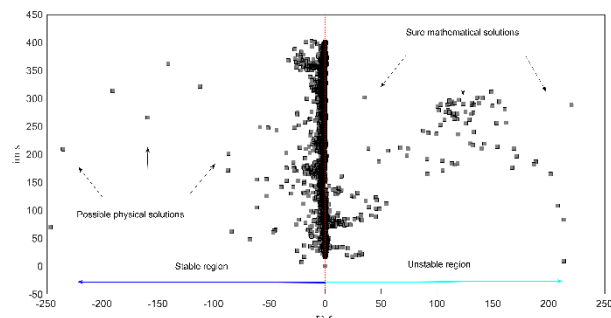


Fig. 3. Mathematical roots.

Fig. 3. shows that some of the mathematical roots show unstable behavior already at the end of the fitting process. The problem is that there are unstable roots hidden on the stable region of the complex plane. This phenomena brings the

question: how can the real, physical solutions be separated from the wrong, mathematical ones.

### 3. Pole iteration

One possible solution candidate could be an iteration process where all of the poles that was calculated in a certain fitting order will be iterated in another fitting order with the Newton-Raphson method. Here, the assumption was that, any physical roots are stable and remain stable through the iteration process. In the other hand the unstable roots may wander to the unstable region of the complex plane through the iteration, thus a filtering condition can be set up to get rid of these solutions.

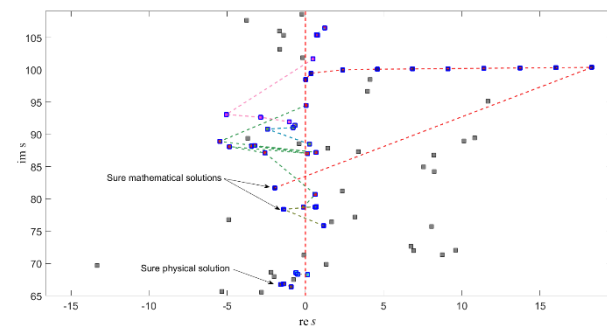


Fig. 4. Pole iteration.

### 4. Conclusions

The fitting process itself is not enough to create and automatic fitting tool, because the appearance of the mathematical solutions. The pole iteration could filter out those solutions but further tests are need to check whether the unstable behavior appears at every mathematical poles during the iteration process. This behavior could serve as a filter condition to separate the solutions.

### Acknowledgement

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# APPRECIABLE CROSS SECTION REDUCTION UPON MODERATE INTRADOS BENDING STRESS INCREASE IN CURVED BEAMS

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## 1. Introduction

A paradoxical behaviour is documented for a straight beam subjected to bending, according to which it is sometimes possible to lower the bending stress by removing material from the zones farthest from the section neutral axis. A rhombic section from which the top and bottom edges are removed is an example, Fig. 1 (a), [1].

Moving to curved beams, it has recently been shown that it is similarly possible to lower the bending stress by laterally removing material from regions close to the neutral axis of the beam section. The rhombic section from which the lateral edges are removed constitutes an instructive example, Fig. 1 (b). This mechanical response too may be classified as paradoxical, since it is possible to simultaneously reduce the bending stress and the beam section area.

However, numerical tests have shown that the stress diminution achievable with this approach is often of the order of an irrelevant 1 per cent, whereas the section area diminution is often in the region of an interesting 10 per cent. To get a more appreciable area reduction, in this paper it was decided to abandon the request for stress diminution, in favour of a less stringent condition on the intrados bending stress, i.e., it is accepted that this stress increases, albeit moderately, upon substantial material removal from the beam section.

This approach employs the classical theory for the bending stress in a curved beam, [2], [3]. A simplifying Gateaux linearization has been adopted. A horizontal strip is defined that delimits the beam cross section zone from which material may favourably be removed.

Preliminary numerical tests have been carried out on a square cross section along whose sides two rectangular grooves are manufactured. The numerical results indicate that by accepting a stress

relative increment of, say, 2 per cent, it is possible to reduce the section area by 20 per cent.

## 2. The paradoxical results

The analytical expression of the bending stress  $\sigma$  for a general radius  $r$  is:

$$\sigma(r) = \frac{M(r_n - r)}{A(r_g - r_n)r} ; \quad r_g = \frac{\int_{r_i}^{r_o} w(r)r \, dr}{A} ; \quad r_n = \frac{A}{\int_{r_i}^{r_o} \frac{w(r)}{r} \, dr} \quad (1)$$

where  $M$  denotes the bending moment,  $r_g$  is the radius of the centre of mass of the beam section,  $r_n$  is the radius of the neutral axis, and  $A$  is the area of the cross section;  $w(r)$  represents the width of the beam section as a function of the general radius  $r$ , and  $r_i$  and  $r_o$  are the inner and outer radii, respectively.

The bending stress is considered in a section described as a composition of the parameters addressing the initial section, endowed with index 0, and of the lateral removal depth  $s(r)$ , Fig. 2. The expression of the bending stress  $\sigma_i$  at the intrados radius  $r_i$ , where the maximum bending stress usually falls, is:

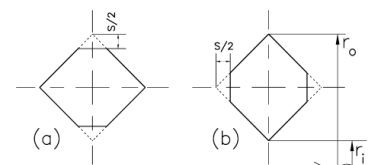


Fig. 1. Examples of paradoxes.

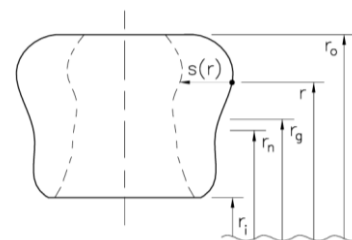


Fig. 2. The main symbols adopted.

$$\sigma_i = M \frac{r_{n,0} - \int_{r_i}^{r_0} s(r) dr - r_i}{\left( A_0 - \int_{r_i}^{r_0} s(r) dr \right) \left( \frac{A_0 r_{g,0} - \int_{r_i}^{r_0} s(r) r dr}{A_0 - \int_{r_i}^{r_0} s(r) dr} - \frac{A_0 - \int_{r_i}^{r_0} s(r) dr}{r_{n,0} - \int_{r_i}^{r_0} \frac{s(r)}{r} dr} \right) r_i} \quad (2)$$

The above equation has been simplified by linearizing the effect of the removal depth  $s(r)$  through a first order Taylor expansion based upon Gateaux differentiation. One obtains:

$$\sigma_i \approx \frac{M(r_{n,0} - r_i)}{A_0(r_{g,0} - r_{n,0})r_i} - \frac{M}{A_0^2(r_{g,0} - r_{n,0})^2 r_i} \int_{r_i}^{r_0} \frac{s(r)}{r} (r_{n,0} - r) [r(r_{n,0} - r_i) - r_{n,0}(r_{g,0} - r_i)] dr \quad (3)$$

Figure 3 presents a trapezoidal cross section, endowed with two lateral grooves with rounded bottom. The  $x$ -variable describes the groove cumulative depth, that quantifies the section reduction; the left  $y$ -variable illustrates the relative stress diminution; the right  $y$ -variable reports the two roots defining the favourable strip from which

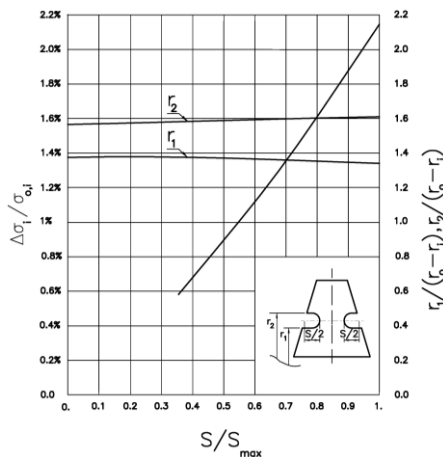


Fig. 3. Trapezoidal section with lateral grooves.

a material removal produces a paradoxical stress diminution. However, this example shows that, for a section diminution of 50 per cent, the stress reduction is a negligible 1 per cent.

### 3. The new results

To get a more appreciable area reduction, in this paper the request for stress diminution is abandoned in favour of a less stringent condition on the intrados bending stress, i.e., it is accepted that this stress increases, albeit moderately, upon substantial material removal. This approach is here quantified for a square cross section in which two rectangular lateral grooves are manufactured. For simplicity, it is assumed that the depth  $S$  of each lateral groove be

described by the expression  $r/[(r_{k,1} + r_{k,2})/2]$ . It is also required that the relative stress increase be proportional to the relative area diminution according to the equation  $\Delta\sigma/\sigma_0 = K\Delta A/A_0$ , where  $K$  is an imposed constant.

The further condition is imposed requiring that the favourable strip width be maximum. With these two conditions, the two roots  $r_{k,1}$  and  $r_{k,2}$  defining the favourable strip may be expressed in analytical form as:

$$r_{k,1}, r_{k,2} = \frac{r_{n,0}(r_{g,0} + r_{n,0} - 2r_i)}{(r_{n,0} - r_i)} + K(r_{g,0} - r_{n,0}) \pm \sqrt{3 \left[ \frac{r_{n,0}(r_{g,0} + r_{n,0} - 2r_i)}{(r_{n,0} - r_i)} + K(r_{g,0} - r_{n,0}) \right]^2 - 12r_{n,0}^2 \frac{(r_{g,0} - r_i)}{(r_{n,0} - r_i)}} \quad (4)$$

### 4. Concluding remarks

For the reference section defined by  $r_i=1$ ,  $r_0=2$ , section width  $b=1$ ,  $K=0.1$  (a relative bending stress increase of 10 per cent of the section reduction is accepted), then  $r_{k,1} \approx 1.31$  and  $r_{k,2} \approx 1.77$ . The width of the favourable strip is therefore 0.46. If the two lateral grooves possess a cumulative depth of 1, the section diminution is 46 per cent (at the cost of a intrados stress increase of 4.6 per cent), but the section is undesirably segmented into two parts. Instead, by retaining a web thickness half the section initial width, the relative area reduction is a still interesting 23 per cent, for a limited stress increase of 2.3 per cent. If the exact equations (1-2) are employed instead of the Gateaux-linearized formula (3), the stress increase becomes 2.6 per cent, while the section reduction is 23 per cent. The linearized model therefore supplies reasonably accurate results.

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# EXPERIMENTAL APPROACH TO THE VERIFICATION OF NUMERICAL MODEL OF HYBRID MACHINE FOR CONSTRUCTION WORKS

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## 1. Introduction

As a result of the growing demand for automating the process of demolishing building structures and for the removal of rocks and spoil, in the mid-nineteenth century, the development of specialized demolition machines took place.

As a result of the insufficient durability of the competing companies' machines, Advanced Robotic Engineering company from Poland decided to construct its own hybrid demolition robot which is shown in the picture below.



**Fig. 1.** Visualization of the electric D-REX demolition machine

In order to verify the correctness of the design assumptions and assess the strength of the robot structure, a strength analysis using the finite element method can be used.

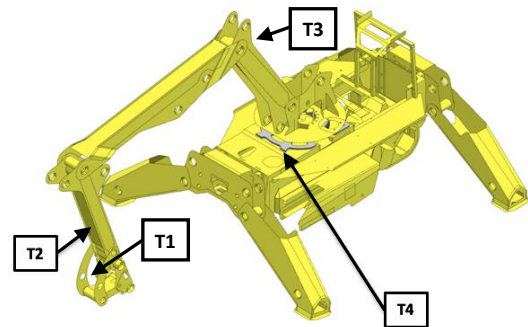
The authors present in the article an experimental approach to verification of numerical model, which will be used to determine the most sensitive points in the construction, where concentration of stresses will occur.

## 2. Experimental studies

Several measurement methods can be used for measurements in the machine's working environment. The most often used method is a strain gauges, and to eliminate measurement

errors in its application resulting from environmental conditions and to determine resonant frequencies a high-speed camera can be used [1]. In this article a strain gauges were used.

In order to determine the mounting points of strain gauges, a preliminary Finite Element Analysis of the arm robot's system was made [2]. Mounting points are shown in Fig. 2. The strain gauge number 4 has been glued in a place where large displacements were observed on the prototype of the machine during its operation.



**Fig. 2.** Mounting points of strain gauges: T – strain gauge

Several load cases of the carrying structures of demolition robot have been defined for which experimental measurements have been carried out. One example of the load of the robot's arm system to verify the numerical model is shown in Fig. 3.

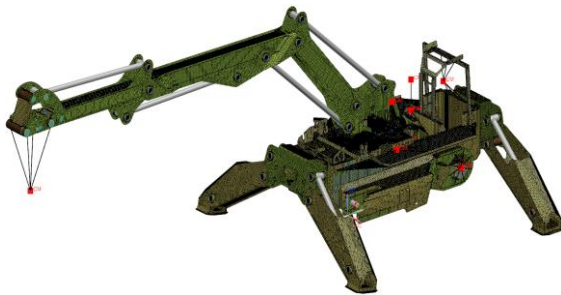


**Fig. 3.** Example of the load of the robot's arm system – steel beams with a total weight  $\approx 48$  kg

The registered signal converted to the appropriate stress values was used to verify the calculation model [3].

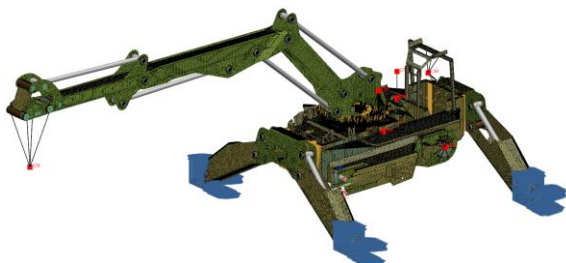
### 3. Numerical model

On the basis of the 3D geometric model of the demolition robot carrying structure, a discrete model with a division into finite elements was built. To build a discrete model solid, thin shell, rod, beam, mass and rigid elements were used. To build the robot structure the thin shell elements were used. Rod elements were used to simulate hydraulic cylinders and beam elements were used to simulate pins. Rigid elements were used to connect individual parts together, taking into account the possibility of moving them in the appropriate axes and to connect the appropriate mass elements to the robot's structure. The robot's discrete model is shown in Fig. 4.



**Fig. 4.** Example of the load of the robot's arm system – steel beams with a total weight  $\approx 48$  kg

On the basis of a discrete model, computational models with appropriate boundary conditions corresponding to measurements on a real object were prepared. There was a rigid contact between the bearing made by solid elements and robot's structure made by thin shell elements. An exemplary calculation model is shown in Fig. 5.

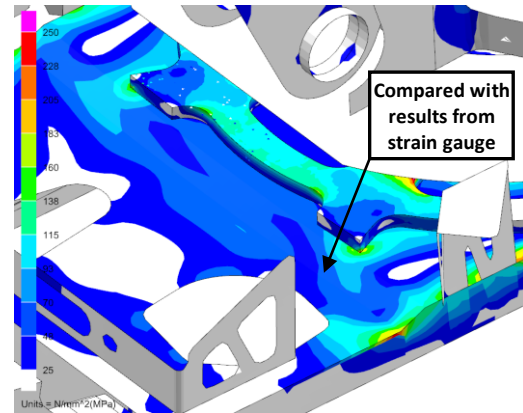


**Fig. 5.** Example of boundary condition – all DOF fixed in support legs

### 4. Results

The results of numerical simulations were used to determine stresses at the places of sticking the

strain gauges on the real object and compare them with the measured values. Exemplary results at the location of strain gauge, in numerical simulation, number 4 are shown in Fig. 6.



**Fig. 6.** Example of the results of numerical simulations – strain gauge number 4

### 5. Summary

Experimental approach to verification of numerical models of the demolition robot with the use of strain gauges allows full adjustment of the numerical model.

The next step in order to check the correctness of the assumptions should be making measurements in the real working conditions of the demolition robot, and in the numerical models, it should be reflected in such a way as to obtain exactly the same stresses in the place of strain gauges as in the case of measurements.

### Acknowledgements

The project was carried with the support of the National Centre for Research and Development in Poland under the “Szybka Ścieżka” program no POIR.01.01.01-00-0582/15-00, in cooperation with Advanced Robotic Engineering Ltd. company.

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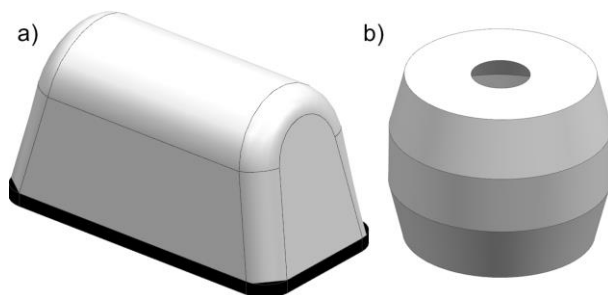
# NUMERICAL AND EXPERIMENTAL ANALYSIS OF THE RUBBER BUMPER STIFFNESS

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## 1. Introduction

Rubber elements are used in heavy duty vehicles as the damping elements or for reducing vibrations caused by the operation of the vehicle. In the suspension systems, they are accompanied by the springs. Their task is to protect the vehicle's frame from the collision with the spring and a possible damage of both elements as a result of high deflection [2]. The rubber element also limits the maximum suspension travel by setting the working range. The selection of the appropriate rubber element and its location on the frame affects the durability of both the spring and the bumper itself. The paper presents the results of the examination of the characteristics of two types of bumpers at different deflection velocities, and then, on the basis of the obtained characteristics, the method of selecting the parameters describing the material from which they were made. The two types of bumpers were considered for the tests: cylindrical and trapezoidal (Figure 1).



**Fig. 1.** Types of the examined bumpers a) trapezoidal, b) cylindrical.

## 2. Experimental testing

Experimental research was conducted on a universal testing machine with 40kN capacity. The tests were performed for two compression

velocities: 0.015m/s and 0.03m/s. Deflection of the cylindrical bumper was equal 38mm, while trapezoidal 30mm. The deformation forms of the bumpers at the maximum deflection are shown in Figures 2 and 3.



**Fig. 2.** Deformation of the cylindrical bumper at maximal deflection.



**Fig. 3.** Deformation of the trapezoidal bumper at maximal deflection.

From the conducted tests, the characteristics of each of the bumpers were obtained. For a trapezoidal bumper (Fig. 4), a clear difference in the characteristics of the bumper between various deformation velocities may be seen. In the case of cylindrical bumper, the difference is visible for the smallest compression velocities, while for other the curves overlap.

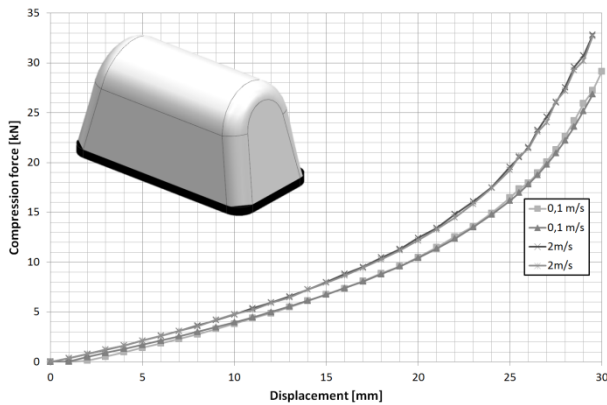


Fig. 4. Characteristic of the trapezoidal bumper.

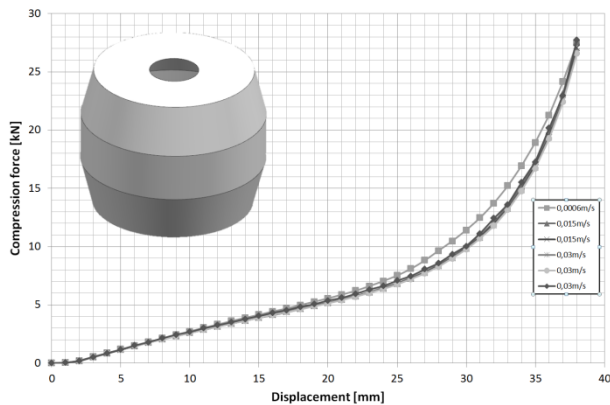


Fig. 5. Characteristic of the cylindrical bumper.

### 3. Model material

The purpose of the numerical analysis was to adjust the characteristics of the bumpers obtained in the numerical examination, to the results achieved from the experimental test. In the analysis a hyperelastic material based on the Mooney-Rivlin model was used [1]. In this model, the behaviour of the material is based on three constants:  $C_{01}$ ,  $C_{10}$  and  $D_1$ . Constant  $C_{01}=0$ , while  $D_1$  is basing on  $C_{01}$  and  $C_{10}$ . The material parameters of both bumpers, determined according to the abovementioned method were used for further analysis of suspension systems, where contact between the spring and the bumpers appear.

Computational models of both bumps with the plates, used during compression test (Fig. 6).

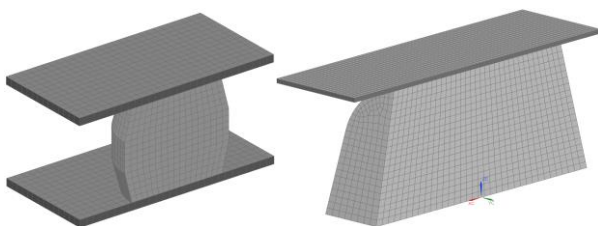


Fig. 6. Discrete models of the bumpers.

On the basis of several numerical analyses, the coefficients describing the rubber material used for modelling of both of the bumpers were selected. Figure 6 shows the contours of the displacements caused by plate's movement.

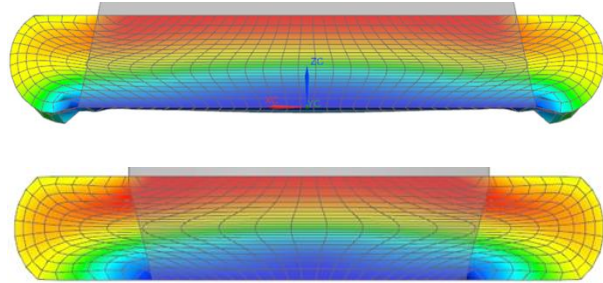


Fig. 7. Contours of the displacement of a) trapezoidal bumper b) cylindrical bumper.

The obtained characteristics were shown in Figure 8. A proper convergence of the results was achieved.

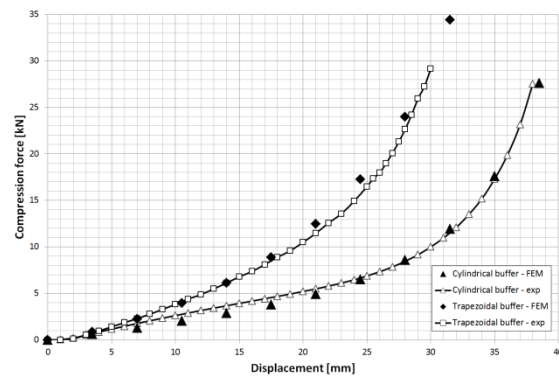


Fig. 8. Comparison of both characteristics.

### 4. Conclusions

The bumpers used in the heavy duty vehicle suspensions transfer the loads in one direction only. The presented method of determining the parameters describing the assumed material model for a unidirectional loads proved compliance with the experiment.

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## IMPACT RESPONSE OF SANDWICH PANELS WITH POLYURETHANE AND POLYSTYRENE CORE AND COMPOSITE FACESHEETS

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### 1. Introduction

Comprehensive monographies on the impact of composite structures [1, 2] cover analytical, experimental and numerical methods and procedures of analyses with laminated or aluminum skins and different types of cores. Experimental determination of the impact behavior of sandwich panels is neither resourceful nor cost effective, and in many cases finite element modeling (FEM) and analysis are used for these types of studies to predict the behavior and failure of these panels. Some of the experimental analyses on sandwich panels [3-5] reveal the great interest in using low density polymeric foams as they have better energy absorption capabilities showing little evidence of debonding at the skin-core interface and as the strain rate sensitivity increases with the density of the core material. Combined experimental and numerical studies in looking to evaluate the response of the sandwich panels constituted a strong option as it was done, only as an example by several researchers [6], and only FEM simulations have attracted more studies as being more convenient; many of them consider a foam core for the sandwich. However, the calibration of the FEM impact model should be carefully done. Only experimental results can provide a good understanding of the impact event and response of the sandwich panel in conjunction to the localized damage produced during low-velocity impact.

### 2. Experimental impact testing

An instrumented Instron Ceast 9340 Drop Tower Impact System used a striker of 20 mm diameter and the impact force was measured during the impact. The initial impact velocity of the striker was measured with an optical cell. The sandwich plates of 140x140 mm were placed on an adjustable in

height test specimen support with a circular hole of 100 mm diameter (Fig. 1), which eventually allowed the striker to fall if the plate was perforated. A clamping ring was pressed over the sandwich plate by a pneumatic system with a maximum force of 3 kN. A special attention was given to the positioning and the alignment of the specimen as to obtain the impact in the middle of the plate. Fig. 1 shows the sandwich panel fixed in between the specimen support and the clamping ring. The energy carrier of gravitationally accelerated type had a mass of 3.15 kg and for most tests two additional masses of 5 kg each were added. Other details were presented in [7]. Data acquisition was done with a frequency of 200 kHz.

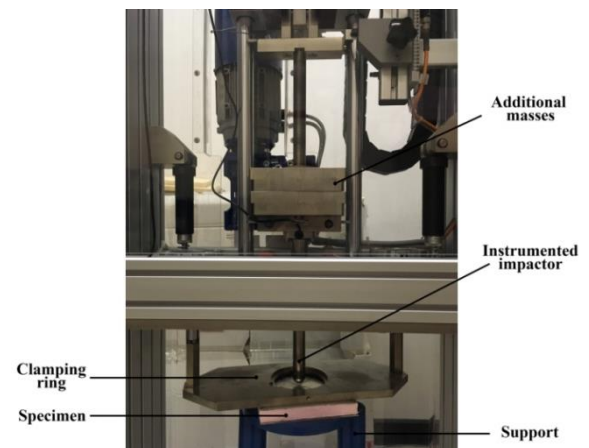


Fig. 1. Impact of sandwich plate during testing.

### 3. Description of sandwich panels

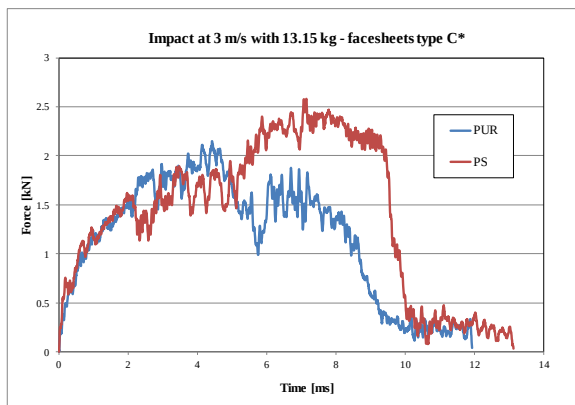
The sandwich panels had skins made of composite facesheets glued with Araldite AW106 (Huntsman) to two types of core: polyurethane (PUR) Necuron 100 of density 100 kg/m<sup>3</sup> and commercial extruded polystyrene (PS) of density 32 kg/m<sup>3</sup> having thicknesses of 12 mm.

The composite facesheets produced at the company Compozite SRL are made of glass fiber roving of 500 grams/m<sup>2</sup>, named RT500, and short glass fibers of 3 mm in different combinations. The epoxy resin is EPOLAM 2017 with the hardener EPOLAM 2017. Facesheets are of five types: A – eight layers of RT500 as reference; B – same as A with an addition of 6.40 grams of short glass fibers at 100 grams of epoxy; C - same as A with an addition of 9.30 grams of short glass fibers at 100 grams of epoxy; B\* - as for B with 6.40 grams of short glass fibers at 100 grams of epoxy without roving; C\* - as for C with 9.30 grams of short glass fibers at 100 grams of epoxy without roving. Thickness of facesheets varies between 3.5 and 5 mm, depending on the type.

The initial velocity of impact was increased from 3 m/s up to 4.5 m/s for both type of panels, as being the maximum speed used in these tests.

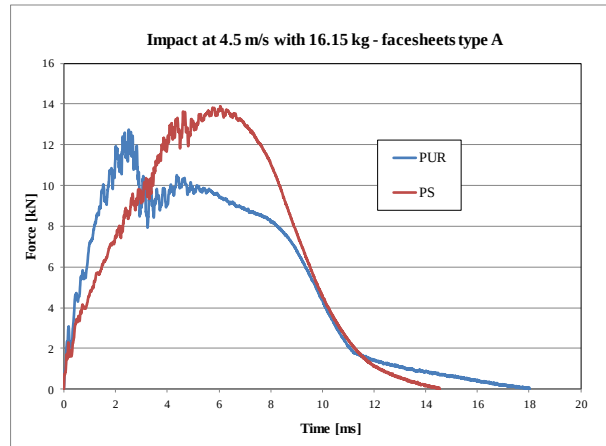
#### 4. Experimental results

At 3 m/s using facesheets type C\* (Fig. 2) the response of the two panels is different, the one with PS core gives a higher contact force during impact and therefore absorbs more of the impact energy. The top facesheet of the PUR panel is damaged more severely. The contact force varies with many fluctuations at this speed as the acquisition frequency is high and local damage of the short fibers is produced.



**Fig. 2.** Contact force variation during impact for C\* facesheets for both type of cores.

Increasing the initial impact velocity to 4.5 m/s and mass to 16.15 kg (Fig. 3) the response of the two types of panels is evidently different. For the PUR foam the response of the panel is more rigid and the top facesheet is almost completely perforated after about 3 ms.



**Fig. 3.** Contact force variation during impact for A facesheets, for both types of cores.

#### 5. Conclusions

The response of the panels is different as the polystyrene core absorbs more efficiently the impact energy regardless the facesheet type.

#### Acknowledgements

This work was supported by a grant from the Romanian National Authority for Scientific Research and Innovation, CCCDI-UEFISCDI, project number 11/2015.

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# MEASURING THE AXIAL FORCE ON TURBO-JET ENGINE ROTOR

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Marko MILOŠ<sup>1</sup>

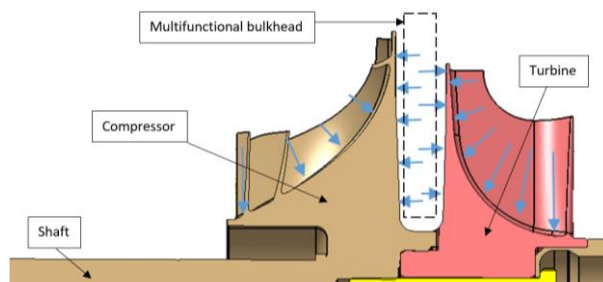
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## 1. Introduction

This paper is focused on measuring the axial force on a turbo-jet engine. The engine operation is based on the Brayton thermodynamic cycle [1]. During this cycle there is interaction between the operating fluid and structure parts. One way of interaction is in the static pressure distribution coming from the compressed operating fluid. This distribution along the surfaces of the rotor assembly (impellers hubs, blades and discs) creates a resultant axial force.

The case study is a relatively small turbo-jet engine of back-to-back rotor concept [2]. This means a radial turbine is installed to drive the radial compressor, placed both on the same shaft and positioned with disk faced to each other, Fig.1. The compressor impeller has a larger diameter and operates on higher values than the turbine impeller so the resultant axial force is very large. This force is the main load of the bearings and it directly governs the bearing's life, so predicting its value is very important even in bearing selection procedure.



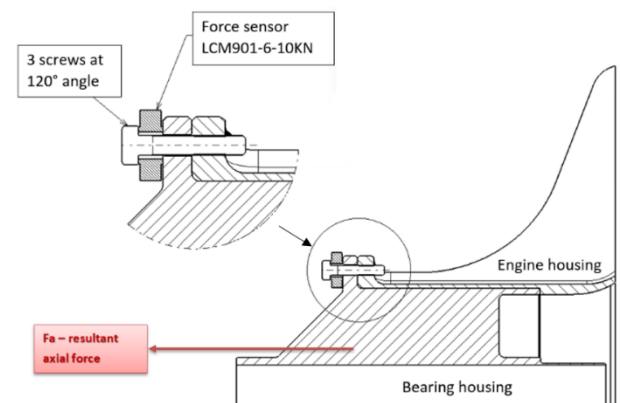
**Fig. 1.** Turbo-jet engine rotor and exposed surfaces to static pressure from the operating fluid

The calculation of the axial force is very difficult because it depends on the geometry of parts, clearances, labyrinth efficiency, rotor rpm, Tesla turbine effect and others. On the other hand there is a multifunctional bulkhead placed between the compressor and turbine which leaves a very narrow cavity where the operating fluid flows. So, there is

no space to install sensors in order to measure pressure values in several points for defining pressure distribution. Very small piezo sensors for high temperature application will be very expensive and out of our reach. So the idea is to measure the resultant axial force indirectly.

## 2. The principle of measurement

This paper presents an innovative way to measure the axial force on the rotor by applying the force sensors indirectly on housing flanges. The solution was to create the bearing housing as a separate part from the rest of the engine housing and to connect them by screws on their flanges, Fig. 2. In this way the screws are exposed to the longitudinal load (axial force).



**Fig. 2.** Principle of measurement and screws position

The axial force that occurs due to static pressure distribution from the operating fluid to the rotor assembly surfaces is carried by the bearings to the bearing housing. The bearing housing is installed in the engine housing along the cylindrical surface with clearance. The clearance is small enough to maintain concentricity of parts and large enough not to make any resistance during sliding of parts. These parts are bonded with 3 screws and the thread is in the engine housing flange, as showed in Fig.2. Based on this, the axial force is carried to the engine

housing with these 3 screws. By installing the force sensors between the bearing housing flange and the screw head the force that tends to separate the flanges can be measured. This way, the axial force presses these 3 sensors and can be calculated as a sum of all measured values from each sensor. For this purpose a sensor LCM901-6-10KN, in the shape of a washer, is used (Fig.3).

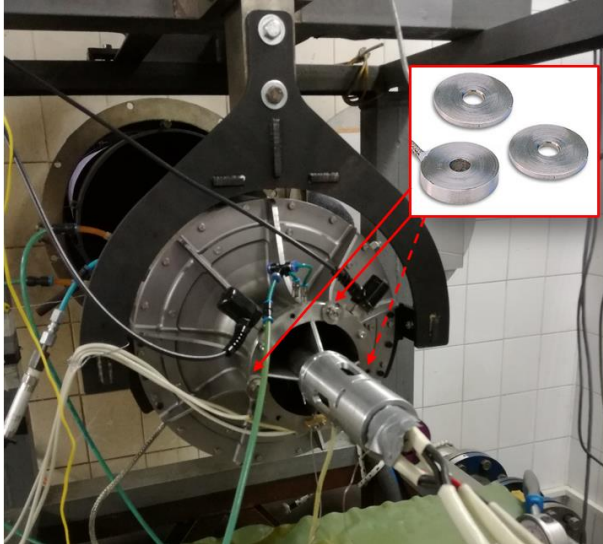


Fig. 3. Measuring system of three sensors installed on the engine

### 3. Experimental measuring

The initial preload of sensors and screws is needed in order to cancel out the clearance in the thread. As the engine starts and accelerates to maximum regime, the axial force increases and the transition phase begins. This phase ends when the axial force becomes bigger than the initial preload. During this phase the axial force stretches the screws and this has a consequence of lowering the preload. But, the sensor on the explained position exhibits and gives the same amount of pressuring force, just the percent of the preload is decreasing while the percent of the axial force increases. When the axial force begins to be larger, the preload and the transition phase vanish and only the axial force is indicated. Based on this, the measuring of the axial force is feasible if the force is larger than the initial preload.

In our case the measuring force is expected to be from 800 N to 1,000 N so we preloaded the screws with only 200 N, approximately 70 N each. This is a very small preload for screws M5, so they are secured by laser welding in several spots on housing flange from opposite side of the screws heads, Fig.2. After that, the calibration was done by applying the

loads in the same direction as the axial force acts in the real test.

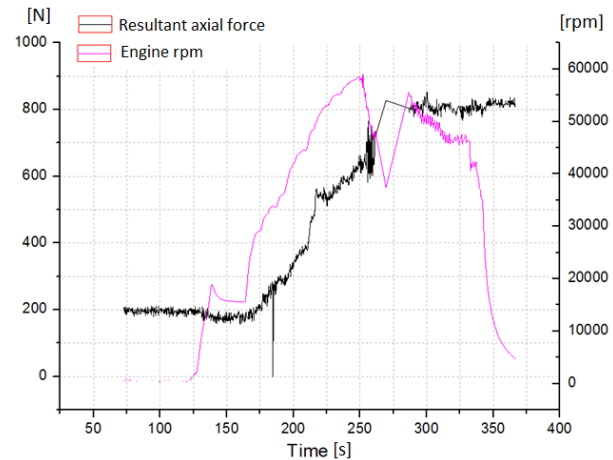


Fig. 4. Resultant axial force in the function of engine rpm

The measured value for the resultant axial force in the function of engine rpm is showed in Fig.4. The sudden drop of engine rpm is an erroneous state of the rpm sensor because of high vibrations. From analyzing the row signal it was concluded that the rpm curve goes in this period up to 62 000 rpm. The sensors and structure after exposure to the force are not very responsive, so the measured values are retained after the sudden shut down for some period of time.

### 4. Conclusions

The paper shows the innovative approach for measuring the axial force on turbo-jet engine rotors. The principle is tested on a real experimental run of turbo-jet engine and concept proof to be feasible. The following work should be directed in the way of changing the sensor type. For example, the force can be measured by conventional strain gauges by determining the elongation of screws.

### Acknowledgements

The authors wish to acknowledge and thank the company EDePro – Engine Design and Production in Belgrade for support and for encouragement in the project realization.

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# METHODOLOGY OPTIMIZATION TESTS OF COMPOSITE ROTOR BLADES OF HELICOPTERS IN EXTREME CONDITIONS

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## 1. Introduction

This paper deals with the methodology of helicopter tale rotor blade analysis, that had been damaged by direct missile hit.

Primarily, the fatigue test of tail rotor blade has been done in order to determine the blade ability to fulfill its vital function. In many cases, fatigue test presents very important, but also final analysis in the design of a blade. The whole process of fatigue test on a test table is based on a document named "Homologation des Fabrications Prevision des Essais Justificatifs" [1, 2, 8].

These analyses were conducted in subsonic wind tunnel Miroslav Nenadovic at the Faculty of Mechanical Engineering in Belgrade.

## 2. The fatigue test of a pierced composite tail rotor blade

In first case, the blade is made out of composite material and has been pierced by one shot from small arms (7.9 mm). Second case was more extreme and the blade made of same material was pierced by three NATO shots (7.62 mm). Even though the blade was badly damaged, analyses have showed that it can function normally. The analyses were conducted on the test table made in laboratories of Faculty of Mechanical Engineering in Belgrade (Fig. 1) [9].

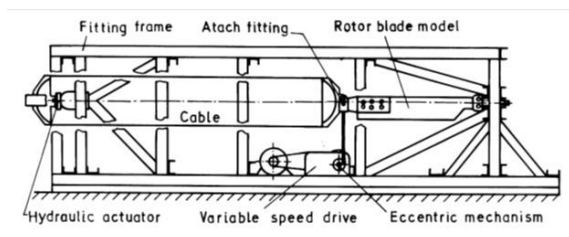


Fig. 1. - Scheme of a test table

## 3. Blade rigidness analysis

The analyses were conducted in the case of undamaged blade, blade pierced by one shot and blade pierced by three shots. Results from three different cases were compared and the following measuring points are presented in Fig. 2.

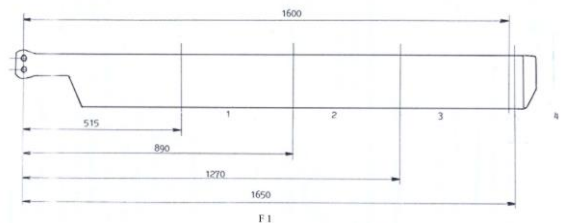


Fig. 2. – Measuring points on a blade

Section one was not considered because it is very near the hobnail and the results in section four are inaccurate because of thinness phenomenon.

## 4. Bump test

The bump test has been conducted in order to analyze the own frequency of a tested blade. It is recommended to use the rubber hammer for frequencies less than 5000 Hz, and to use plastic or wooden hammer for frequencies higher than previously mentioned limit. The rubber hammer was used for the analyses described in this paper. HBM accelerometers put on every third of a blade and SPIDER 8 acquisition system were deployed on a blade [3, 7].

## 5. FFT analysis

FFT analysis (Fast Fourier Transformation) presents an algorithm that helps computer to conduct faster discrete Fourier transformation [4, 5, 6]. In Fig. 3. are showed bump test results for 80800 N load after on shot.

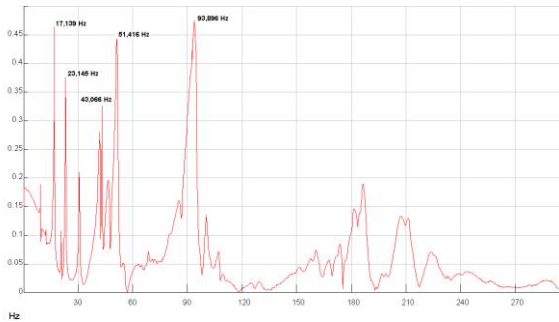


Fig. 3. – Bump test results for 80800 N load after one shot

## 6. Conclusions

The function of tail rotor blade on a helicopter is a very complex technical problem. In order to understand and simulate the functioning of a blade in real conditions, special test table for blade seizure and analysis was created. Analysis of composite blade done by previously mentioned methodology has shown that the composite blade can function normally even with three big damages. This methodology could be successfully used for blades made of different materials and with different damage extent, thus giving relevant qualitative assessment.

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# EXPERIMENTAL STRESS ANALYSIS OF A CANTILEVER BEAM

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## 1. Introduction

The experimental stress analysis is subject of research in papers [1-4]. In this paper we aim to determine the normal stress, existing in a beam of length  $L_0$  and thickness  $h_0$  (position 3, fig. 1), fixed at one end and loaded at the other end by a force  $G$ , whose magnitude increases progressively. The following methods will be used to find stress:

-Analytical method, which allows the evaluation of the normal stresses in equally bending resistance and constant sectional beams, depending on the  $x$ -positions of the analyzed sections.

- The method of determining stresses by experimentally measuring deformations using resistive electrical transducers. This applies to beams of the same type, depending on the  $x$ -positions of the analyzed sections.

## 2. Experimental setup

It is considered a beam of constant length  $L_0$  [mm] and thickness  $h_0$  [mm] (fig. 1), fixed at one end and loaded at the other end by placing the masses  $m$  [kg] whose size increases progressively. In every transverse section at the distance  $x$  from the point of actuation of the force, a bending moment  $M_i$  acts.

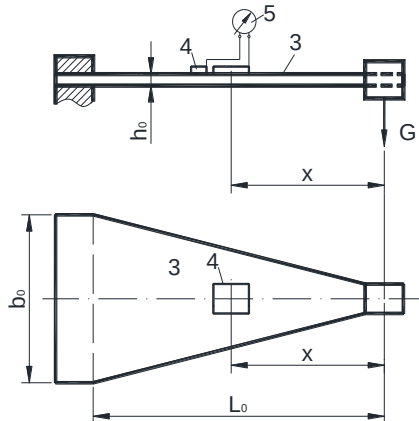


Fig. 1. Equal strength beam.

$$M_i = 9,81 \cdot m \cdot x [Nmm] \quad x \in [0, L_0] \quad (1)$$

Bending generates a distribution of normal stress characterized by the theoretical maximum value:

$$\sigma_{\max}^t(x) = \frac{M_i(x)}{W_y(x)} [MPa], \quad x \in [0, L_0] \quad (2)$$

$$\text{where: } W_y(x) = \frac{b(x) \cdot h_0^2}{6} [mm^3]; \quad x \in [0, L_0]$$

In this paper a beam of equal strength is tested. It has a triangular profile and its current width varies linearly according to the law:

$$\sigma_{\max}^t(x) = \frac{M_i(x)}{W_y(x)} [MPa], \quad x \in [0, L_0] \quad (3)$$

$$\text{Results: } W_y(x) = \frac{b(x) \cdot h_0^2}{6} [mm^3]; \quad x \in [0, L_0]$$

In that case, the theoretical stress is computed with Eq. (4):

$$\begin{aligned} \sigma^t(x) &= \frac{G \cdot x}{\frac{b(x) \cdot h_0^2}{6}} = \frac{6 \cdot G \cdot x}{\frac{b_0}{L_0} \cdot x \cdot h_0^2} = \\ &= \frac{6 \cdot G \cdot L_0}{b_0 \cdot h_0^2} = \text{const.} [Mpa] \end{aligned} \quad (4)$$

The normal stress is:

$$\begin{aligned} \sigma_{\max}^t &= \sigma^t(x) = \frac{6 \cdot G \cdot L_0}{b_0 \cdot h_0^2} = \\ &= \text{const.} [Mpa] \end{aligned} \quad (5)$$

And the specific deformation is:

$$\varepsilon = \frac{6G \cdot L_0}{b_0 h_0^2 E} \quad (6)$$

The second method involves, determining the stress by experimentally measuring deformations,

using resistive electrical transducers. This time, two resistive electric transducers (position 4), which are fixed at the location  $x$  from the application point of the load  $G$  on the upper and lower surfaces of the beams, will be able to measure by means of the MGCPlus data acquisition system, the specific elongations  $\varepsilon_{\max}^{\text{ex}} = \Delta x / L_0$  existing in that cross-section.

In each section  $x$  two transducers were mounted, one on the top and one on the lower surface, the first with the grid along the longitudinal beams of the beams, and the second transverse. These two are mounted in semi-Wheatstone bridge connection because the assembly is insensitive to parasitic effects. Through the transducers, the acquisition system provides the experimental values of stress:

$$\sigma_{\max}^{\text{ex}}(x) = E \cdot \varepsilon_{\max}^{\text{ex}} \cdot 10^{-6}; x \in [0, L_0] \quad (7)$$

where:  $E$  [MPa] is the modulus of elasticity of the material used. For the application of the methods described above, the device shown in Fig. 2, is used. It consists of the base plate 1, the support arm 2, the part 3 analyzed (the resistance beam of Fig. 1), the resistive electric transducers 4, the measuring device MGCPlus 5 and the mechanical loading system 6.

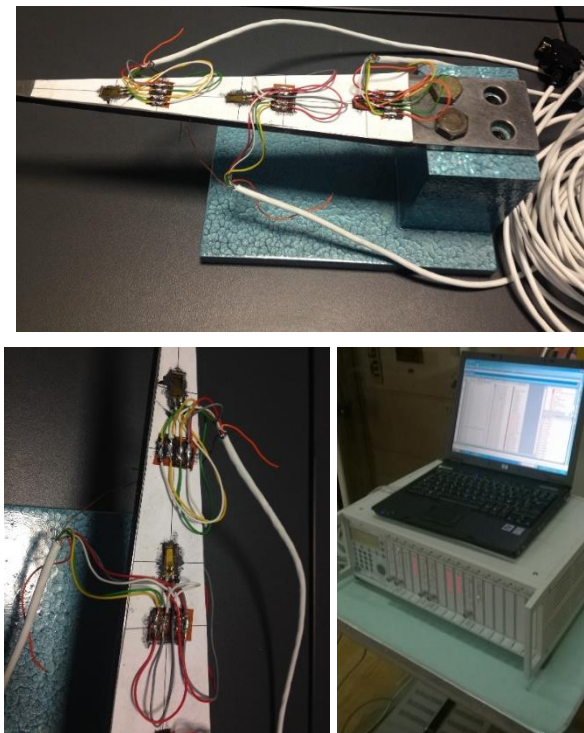


Fig. 2. Experimental setup.

### 3. Experimental results

The obtained experimental results are presented in Table 1.

Table 1. Experimental results

| Case no. | Force [N] | Stress [MPa] | Specific deformation [ $\mu\text{m/m}$ ] |
|----------|-----------|--------------|------------------------------------------|
| 1        | 10        | 6,1          | 12,4                                     |
| 2        | 20        | 12,3         | 24,2                                     |
| 3        | 30        | 20,2         | 36,6                                     |

### 4. Conclusions

The geometry of the profile of the beam, represented in Fig. 2, is expressed analytically by Eq. (2). According to the relation (4), the maximum normal stress value is identical in each cross section of the beam. In order to verify this result, the analytical method and the strain gauge method for the determination of the stresses by experimental measurement of the deformations will be used. Both have been explained above. To apply the analytical method, it is necessary to measure the geometrical characteristics,  $h_0$ ,  $b_0$  and  $L_0$  of the beam. Instead, for the application of the experimental method, a location  $x$  [mm] is chosen arbitrarily from the point of application of the load  $G$ , in which two identical strain gauge transducers are fixed by gluing on the beam surfaces, which are connected in a Wheatstone connection assembly to the MGCPlus acquisition system.

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## **SECTION II: MATERIALS CHARACTERIZATION AND TESTING**



# EXPERIMENTAL DYNAMICAL ANALYSIS OF SPECIMENS' MATERIAL PROPERTIES MANUFACTURED BY ADDITIVE TECHNOLOGIES

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## 1. Introduction

In the recent years 3D printing attracted a lot of attention thanks to the rapid development of the technologies. At the same time the need for a good alternative of the currently popular non-biodegradable plastics also arise. Polylactic acid (PLA) seems to be a good option replacing other plastics. With additive manufacturing technologies using biodegradable plastics, which become a viable option for rapid prototyping or custom manufacturing, we need a deep understanding of the material properties [1] [2]. We looked at the available papers on PLA's material properties. There are only a few about modeling the material properties and the results show big deviation regarding the same properties [3] [4] [5]. Our goal is to better know these parts' damping properties and the effects of the printing parameters on material properties, more precisely the printing direction.

## 2. Materials and methodology

After studying the available literature, we were certain that the positioning of the part in the printer has a massive impact on the material properties. We found that the tensile strength in the printing direction was 50% less than in the other two directions [3] [4]. This coincides with our assumption: parts made by this method show orthotropic properties [5]. It is clear that the main factor is the printing direction therefore we focus on that printing property.

To keep our model simple, we print our specimen as prismatic columns.

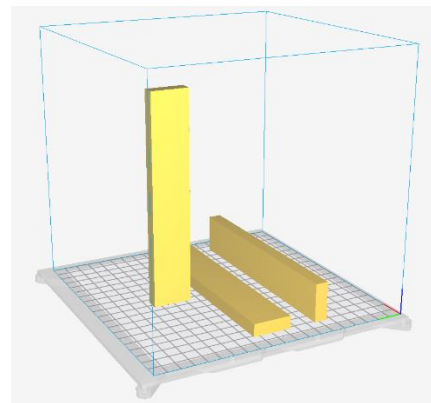


Fig. 1. Printing directions of the specimen. (A, B and C)

For every direction, we printed 3 specimens to check the reproducibility and eliminate errors of the printer. So far, the two laying positions have been tested. We tested the parts as cantilevers. For excitation we used an impact hammer, and measured the response using accelerometers. We located the specimen's natural frequencies (bending and torsion), damping ratios and mode shapes in the  $f=0-1000$  Hz range. For every mode we checked if the mode shape is *classical normal mode* or not. As known classical normal modes' mode shapes show standing wave nature, equivalent to the undamped system's mode shapes. Thus, the mathematical model is significantly simpler.

We also separated the detected mode shapes into two groups depending on the nature of the mode shape: bending and torsion. To avoid false results, to detect natural frequencies, we also measured the base's natural frequencies. To identify the damping mechanism, we will define regression functions which express the damping factor's frequency dependence [6]. These functions have a direct connection to the damping factor's function. For the measured Frequency Response Function's (FRF from now) analytic approximation we used the partial fraction form:

$$H(j\omega) = \sum_{i=1}^n \left[ \frac{P_i}{j\omega - \lambda_i} + \frac{\overline{P_i}}{j\omega - \overline{\lambda_i}} \right] \quad (1)$$

$$\lambda_i = -\sigma_i + j\omega_{d,i} \quad (2)$$

where:  $P_i$  are residuum of mode  $i$  [m/sec/N],  $\lambda_i$  are the complex eigenvalue of mode  $i$  [rad/s],  $\sigma_i$  are modal damping of mode  $i$ ,  $\omega_{d,i}$  are damped natural frequency of mode  $i$ , and  $j$  is the imaginary part.

### 3. Results

Fig. 2 shows the *FRF* of a part printed with method “B”. In Fig. 3 four dominant modal circles can be seen. Further analysis has to be made to estimate the modes with smaller amplitude (near the origin).

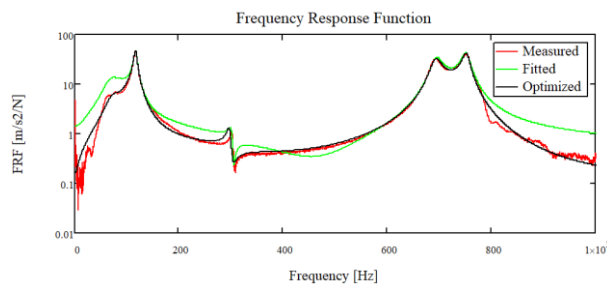


Fig. 2. FRF of part B1

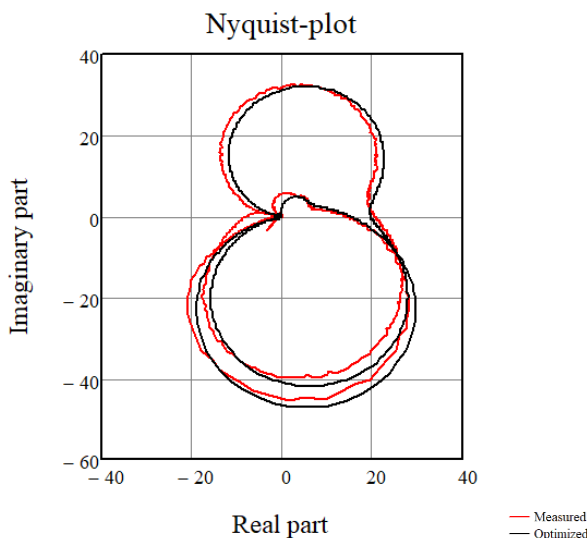


Fig. 2. Nyquist-plot of part B1

Using this method, we could detect 4 natural frequencies (after eliminating the base's modes). For the classification we use the *Nyquist-plot* of the *FRF*.

Table 1. Classification of the mode shapes

| Mode $i$    | 1      | 2      | 3      | 4      |
|-------------|--------|--------|--------|--------|
| $f_i$ [Hz]  | 116,87 | 304,58 | 692,03 | 762,96 |
| $\xi_i$ [%] | 2,61   | 9,14   | 1,49   | 0,93   |
| Type        | B      | T      | T      | B      |

### 4. Conclusions, further research

The results show little difference between the two laying parts (B and C). With the last measurements done we will be able to compare the different orientations. When processing the data, we suspected non-linearity of the material properties. Further research has to be done to fully identify the proper material model and to detect more mode shapes for defining the regression functions.

### Acknowledgements

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# INVESTIGATION OF THE SHRINKAGE BEHAVIOR OF CONCRETE CONSIDERING THE MOISTURE CONTENT

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## 1. Introduction

As a consequence of the increasing traffic load and the progression of the mean age of bridges, measures of rehabilitation are becoming more and more important. One such measure is adding a concrete overlay. Therefore, the aim of this study is to provide a better understanding of the material parameters related to shrinkage and to determine the respective parameters for multiphase modeling of concrete structures.

## 2. Material and Methods

In the following, concrete specimens at different ages with (compressive) strength class C30/37 are investigated.

First of all, the development of the uniaxial compressive strength and the Young's modulus are determined and compared to the suggested evolution in EC2 [1] and MC10 [2].

Furthermore, the hygric parameters are determined. The desorption isotherms as well as the adsorption isotherms are identified on 8 mm thick concrete slices. The latter were obtained from a two-year-old, sealed specimen by wet sawing. Subsequently, these thin slices are stored in desiccators at relative humidities (RH) of 98, 85, 75, 59, and 43%, at 20±1°C and are observed until, according to EN ISO 12571 [3], constant mass is achieved.

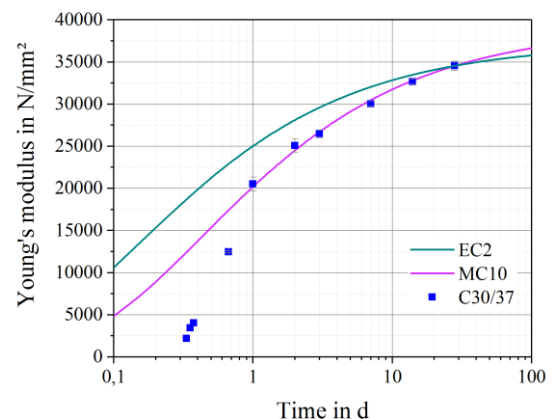
Moreover, the evolution of the autogenous shrinkage strain, the pure drying shrinkage strain, and the combined autogenous and drying shrinkage strain are investigated on thin concrete slices, stored in desiccators. To measure the shrinkage strains each slice is equipped with eight measuring discs. Two discs form one measuring length of 100mm, resulting in four measurement distances.

The autogenous shrinkage strain is determined on sealed specimens from the age of one day,

whereas the drying shrinkage strain is measured on two-year-old specimens, which therefore can be assumed as completely hydrated. Likewise, the combined autogenous shrinkage and drying shrinkage strain is determined using specimens from the age of one day, which are simultaneously hydrating and drying. Moreover, the influence of the concrete age on the combined autogenous and drying shrinkage strain is analysed.

## 3. Results

First, the time-dependent increase of the Young's modulus is shown in figure 1 on a logarithmic time scale.



**Fig. 1.** Development of Young's modulus

It can be seen, that the EC2 overestimates the material stiffness, whereas the MC10 shows good agreement with the experimental data. However, none of the two models considers the low stiffness up to the age of one day.

Furthermore, the desorption isotherm (DIT) and adsorption isotherm (AIT) are shown in figure 2. The first point on the desorption isotherm refers to a RH of 85%, since the internal RH of two-year-old specimens for a comparable concrete is 93% according to [4]. The test results also show a

considerable hysteresis between the drying and the wetting path.

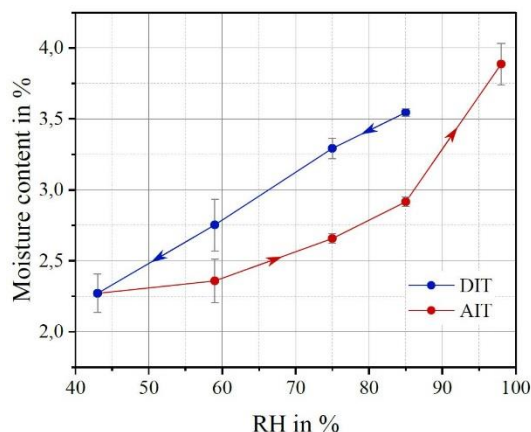


Fig. 2. Desorption and adsorption isotherms

In addition, the autogenous shrinkage strain is measured on sealed specimens. The mean value of the ultimate autogenous shrinkage strain is about -0,09‰ (after 100 days) with a standard deviation of 0,038‰ (measuring accuracy: 0,03‰). The specimens used for the shrinkage tests are named according to the following code: DIT\_age(in days)\_RH.

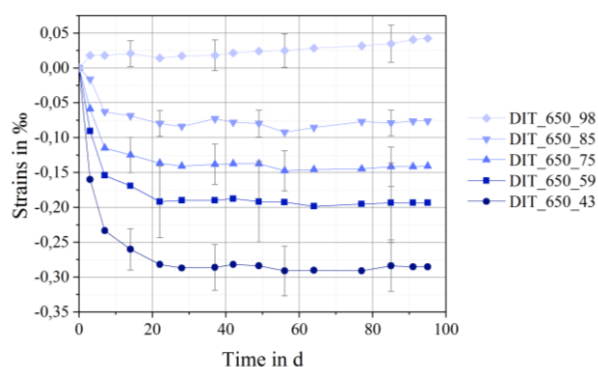


Fig. 3. Drying shrinkage strain

Moreover, the pure drying shrinkage strains for different ambient conditions are plotted in figure 3. Firstly, it can be seen, that for RH of 98% the concrete is swelling, which implies the internal RH is below 98%. This confirms the internal RH of concrete at the respective age provided in [4]. Secondly, the shrinkage strain increases in absolute values with decreasing RH.

Figure 4 illustrates the mean values of the combined autogenous and drying shrinkage strain at a RH of 59%. The drying process starts at ages of 1, 2, 7, 22, and 650 days after casting. However, the result of the 650-day-old specimens represents the pure drying shrinkage strain and is included for comparison with the combined autogenous and

drying shrinkage strain. Nevertheless, it can be seen that drying at early ages has a huge impact on the ultimate shrinkage strain. With increasing age at exposure to drying, the ultimate shrinking strain decreases, approaching the pure ultimate drying shrinkage strain.

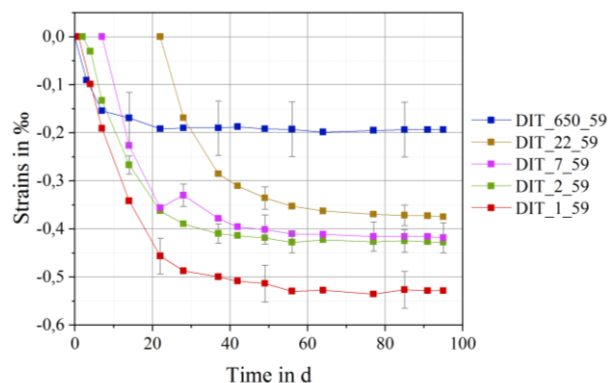


Fig. 4. Combined autogenous and drying shrinkage strain

## 4. Conclusions

In the present study the time dependent evolution of (i) the compressive strength, (ii) the Young's modulus, (iii) the autogenous shrinkage strain, (iv) the drying shrinkage strain and (v) the combined autogenous and drying shrinkage strain were investigated. Furthermore, (vi) the desorption and adsorption isotherms were determined. The collected data can now be used for calibration of material parameters required in coupled multiphase models of concrete.

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# DAMAGE OBSERVATION IN GLASS FIBER REINFORCED COMPOSITES VIA M-TOMOGRAPHY

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## 1. Introduction

The increasing use of fibre reinforced composites as structural components yielded an increased interest in identification and/or validation of their mechanical behaviour. Furthermore, an extensive research on composite materials has been focused to determine damage initiation and growth that induces failure.

Although observations of the composite behaviour based on 2D digital image correlation (DIC) and stereocorrelation have been widely used in the past, the development of laboratory μtomography equipments enable for the visualisation of the microstructure in the bulk of the specimen [1]. Furthermore, the output of tomography scanning (*i.e.* 3D reconstructed volumes) can be used for correlation techniques in order to measure displacement fields over the entire volume of interest.

The purpose of experimental work conducted herein was to apply cyclic uniaxial loadings and monitor damage development via μ-tomography. Global Digital Volume Correlation (DVC) was used to measure displacement and strain fields. Furthermore, correlation residuals allowed damage mechanisms occurring within the glass fibre/epoxy resin composite to be detected.

## 2. Experimental procedure

### 2.1 Material and Methods

The experimental investigation was performed on glass fibre reinforced epoxy resin composites. The R-glass fibre mat with an areal density of 300 g/m<sup>2</sup> and 50 mm long fibres was used as the reinforcement in laminae. The epoxy resin mixture was applied using a roller and a brush, and the laminate was cured under pressure for 24 h. After the moulding process, post-curing was conducted at 100° C for 3 h. The proposed procedure was applied

to prepare composite plates with dimensions of 300 × 300 × 5.6 mm.

The conducted uniaxial experiments were monitored with one CCD camera and the acquired series of images was analysed via global 2D DIC in order to estimate the material response under prescribed load. Moreover, the reconstructed volumes (*i.e.* discrete scalar matrices representing grey levels determined by the microstructure absorption of X-rays) were analysed in order to assess displacement and strain fields in the bulk of the dog bone specimen. The DVC technique used in this work is an extension of 2D global DIC and it consists of minimising the L2-norm of the field of correlation residuals

$$\varphi(\mathbf{x}) = \left| f(\mathbf{x}) - g(\mathbf{x} + \mathbf{u}(\mathbf{x})) \right| \quad (1)$$

where  $f(\mathbf{x})$  is the reference volume,  $g(\mathbf{x})$  the deformed volume, and  $\mathbf{u}(\mathbf{x})$  the sought displacement field. The main assumption of the proposed algorithm is that the displacement field is continuous over the region of interest. The displacement field is expressed as

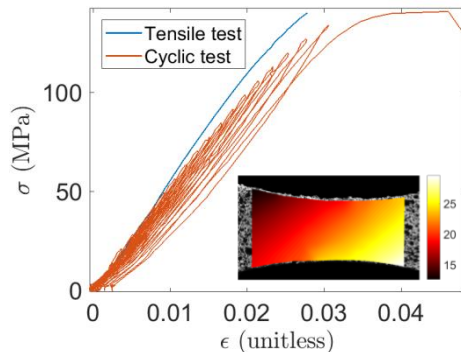
$$\mathbf{u}(\mathbf{x}) = \sum_n u_n \psi_n(\mathbf{x}) \quad (2)$$

where  $\psi_n$  are (chosen) vector fields, and  $u_n$  the associated degrees of freedom.

### 2.2 Tensile and cyclic experiments monitored with 2D DIC

In order to estimate the global material behaviour two classical experiments (*i.e.* uniaxial monotonic and cyclic tension) were carried out on an electromechanical testing machine. The tests were performed on dog bone samples with a radius equal to 48 mm, and a ligament width of 6 mm. The material behaviour from the two experiments is shown in Fig. 1. The Young's modulus is such that  $E = 5.6$  GPa, and the ultimate tensile strength  $\sigma_m = 140$  MPa. For the second mechanical test,

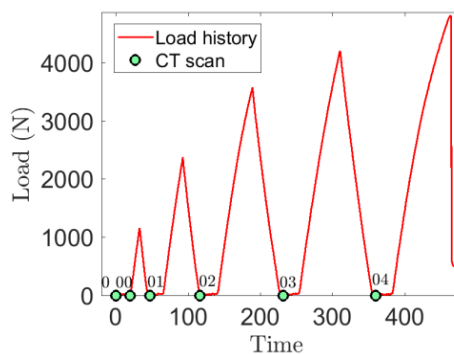
cyclic loading was prescribed in order to evaluate damage growth. The loading and unloading were incrementally increased by 250 N per cycle. A total of twenty cycles was applied.



**Fig. 1.** Stress strain curves obtained from monotonic and cyclic uniaxial tension

### 2.3 Cyclic ex-situ tomography experiment

From the analysed cyclic experiment it was decided to acquire five scans with the maximum load corresponding to 25%, 50%, 75% and 90% (Fig. 2.) of the ultimate tensile strength identified previously. The microtomographic ex situ experiments were performed using the X50+ computed tomograph (North Star Imaging). Before performing the first mechanical loading cycle on the uniaxial testing machine, 3D scans were acquired in order to assess the standard displacement and strain resolutions.



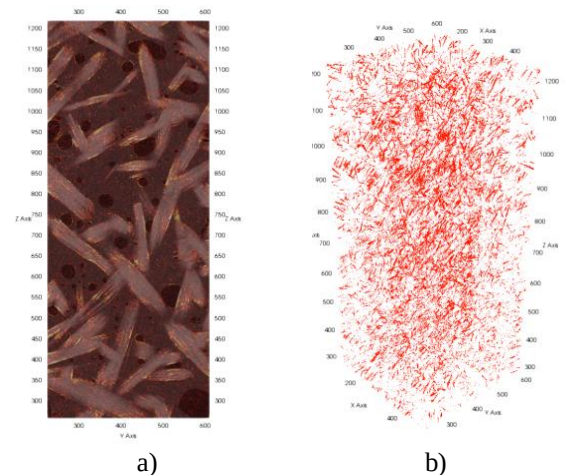
**Fig. 2.** Loading history of the ex situ experiment with the corresponding scans.

For each prescribed loading and unloading step the experiment was monitored via 2D DIC in order to evaluate the global strain levels. When the unloaded step was reached, the specimen was unmounted from the testing machine and X-ray scans were performed.

## 3. Results and Conclusion

Each tomographic scan was initially acquired with a definition of  $1447 \times 1446 \times 1789$  voxels,

whose physical size is  $10.9 \mu\text{m}$ , encoded as 8-bit deep grey levels. In those images, a region of interest was defined in which the displacement field was measured. The size of the ROI was adjusted to the sample geometry focusing on the gauge volume of the sample, whose size was  $384 \times 384 \times 960$  voxels. C8-DVC [2] with 96 voxel elements was used to evaluate the displacement and strain fields between the undeformed volume (scan 0) and deformed ones (scans 00-4). Since, large element sizes were used, the correlation residuals (Fig. 3.) enable damage accumulation to be observed [3]. The correlation residuals detect discontinuities in the displacement fields, which can be interpreted as damage.



**Fig. 3.** Grey level residual maps corresponding to the a) y-z plane axis and b) over the entire volume of interest for scan 04.

Overlaying the correlation residual map on top of the corresponding microstructure (Fig. 3(a)) points out higher residuals around glass fibres. Furthermore, in Fig. 3(b) multiple micro-damage branches are detected, which implies that dominant macrocracks are not formed at 90% of the maximum load.

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## CHARACTERIZATION OF A CORK-RUBBER COMPOSITE USING ADVANCED MATERIAL MODELS

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### 1. Introduction

Cork-rubber composites are modern materials belonging to a group of composition cork. These composites are largely used as a dumping and insulation layers in hybrid composite structures. Composition cork materials are made from cork particles bonded together using several types of adhesive agents. Investigated composite material Amorim ACM87 consists of cork particles bonded together using synthetic rubber (Fig. 1).

The behaviour of the material is very complex. The material behaviour cannot be described using simple elastic models and small strain theory in comparison to metals. Advanced viscoelastic/viscoplastic material models must be introduced to describe the behaviour of cork rubber composites. Based on earlier research, it was proved that the response of the material shows dominant viscoelastic effect with high strain rate dependence [1].

Parameters of advanced material models can be obtained using a material parameter calibration routine. The method is based on a numerical optimization to find the minimum difference in between the experimental data and the model predictions. The aim of this contribution is to compare the performance of different advanced viscoelastic/viscoplastic models used to describe the material behaviour of the investigated Amorim ACM87 composite.

### 2. Experiments

Experimental data (stress-strain curves) are inputs of the material parameter calibration. Behaviour of the material cannot be determined properly using single uniaxial or shear experiments.

In addition, the parameters calibrated using a single type of experiment will probably not be suitable for any other loading mode.

Series of experiments were performed and discussed in [1] and [2]. The behaviour of the material was investigated under various different loading modes and strain rates. Tests in uniaxial tension, biaxial tension, uniaxial compression, volumetric compression, simple shear and arcan shear were performed to obtain relevant information about the characteristics of the material. These data are used for the calibration.

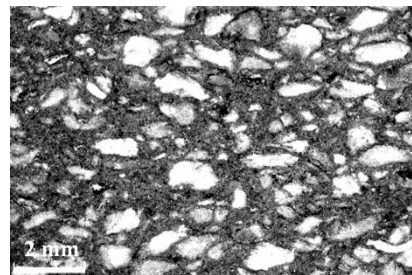


Fig. 1. Cork composite ACM87 structure.

### 3. Material parameters calibration

The process of parameters calibration of advanced viscoelastic/viscoplastic models is a very complex task. It requires experimental data in proper form, numerical simulations corresponding to experiments and an optimization task to identify the best set of parameters. The optimization is based on minimization of a difference between stress-strain curves taken from experiments and numerical simulations.

A combination of finite element method and optimization software can be used for the material parameters calibration as shown in [4]. This

approach is versatile, however, it requires lot of time and licenses to calibrate parameters. Software MCalibration is a special tool designed for material parameters calibration. It offers different advanced user-material models including those provided in the PolyUMod library [5]. In this software, all necessary functions are implemented and large material library with advanced material models is included. We used this software in the present contribution.

#### 4. Material models

Advanced viscoelastic/viscoplastic models are suitable for modelling of non-linear, time dependent, large strain behaviour of viscoelastic materials [3]. The models can be represented using serially or parallel connected elastic, viscous, plastic, and other elements as illustrated in Fig. 2.

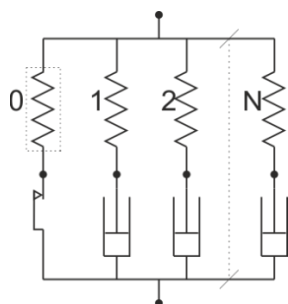


Fig. 2. Rheological representation of Parallel Network Model.

One of the most advanced material model for polymers is the Parallel Network Model (PNM, Fig. 2). This model allows to combine a large number of elastic (hyperelastic), viscous and other material models in one framework. The parameters calibration of the model becomes more complicated with increasing number of networks. Special cases of PNM are dual and three network models for instance.

The Bergström-Boyce (BB) model, despite its simplicity, proves very good capability for characterizing the behaviour of elastomers and bio materials [3]. This model was also used for the calibration in this work.

#### 5. Bergström-Boyce model calibration

Material parameters of the BB model were calibrated using MCalibration. The behaviour in uniaxial tension, uniaxial compression, and simple shear was simulated and compared with experimental data. Investigated strain rate was  $0.01\text{s}^{-1}$  and maximal engineering strain was 10%.

One example of the comparisons of stress-strain curves is shown in Fig. 3. The curves correspond for all of the loading modes used in the calibration.

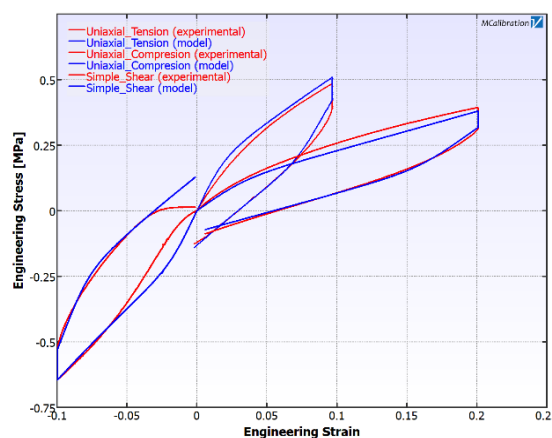


Fig. 3 Stress-strain curves experiments and predictions.

#### 6. Conclusion

The Bergström-Boyce material model can be used for characterizing the behaviour of the cork-rubber composite Amorim ACM87. The model was validated for maximal engineering strains of 10% and strain rates of  $0.01\text{s}^{-1}$ .

#### Acknowledgements

The work has been supported by the grant project SGS-2016-038 and by the UNKP-17-4-III New National Excellence Program of the Ministry of Human Capacities, Hungary.

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# YIELDING AND STRAIN LOCALIZATION EFFECTS IN GUM METAL - A UNIQUE TI ALLOY - INVESTIGATED BY DIGITAL IMAGE CORRELATION AND INFRARED THERMOGRAPHY

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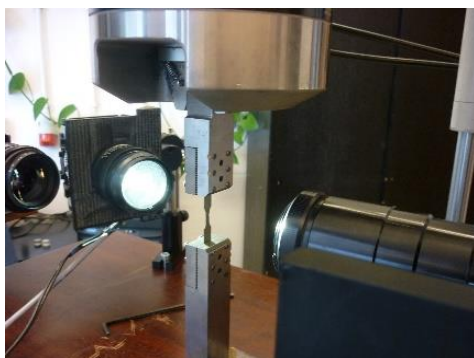
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## 1. Introduction

The research concerns investigation of yielding and strain localization phenomena in a  $\beta$ -Ti alloy characterized by unique elastic-plastic properties, named Gum Metal [1,2]. Digital Image Correlation (DIC) and Infrared Thermography (IRT) were used [3]. The Gum Metal was subjected to tension on testing machine (TM) at three various strain rates up to rupture. Displacement and strain distributions were determined on the basis of DIC algorithm. The related temperature variations were found in contactless manner using IR camera. Mechanical and the corresponding thermal data were used to study large nonlinear reversible strain of the alloy.

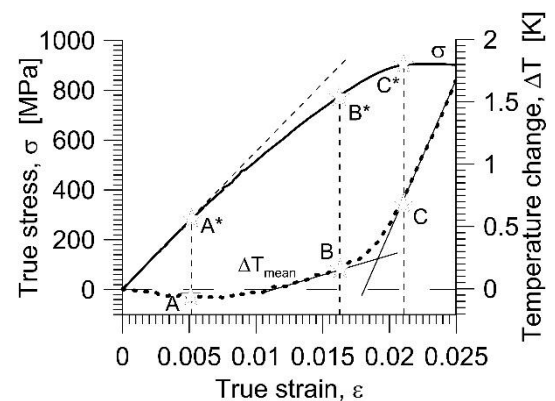
## 2. Experimental results

The setup [Fig. 1] consists of MTS 858 TM and two cameras working in two different spectral ranges: visible (0.3-1  $\mu$ m) Manta G-125B and IR (3-5  $\mu$ m) ThermoCam Phoenix. Dog bone sample with sizes 100 x 8 x 0.5 mm were used, gauge part 7 x 4; length of virtual extensometer for DIC - 7 mm. More experimental data for investigation of effects of thermomechanical couplings in Gum Metal under tensile loading is presented in [3].



**Fig. 1.** Photograph of experimental set-up used for investigation of deformation and thermal fields

The stress  $\sigma$  and mean temperature changes ( $\Delta T_{mean}$ ) of the Gum Metal specimen vs. strain, obtained during tension at strain rate  $10^{-2}s^{-1}$  up to strain 0.025, is shown in Fig. 2. The elastic yield limit, corresponding to maximal drop in the specimen temperature, and limit of mechanically reversible deformation, found in subsequent cyclic loading [3], are marked by A\* and B\*, respectively.



**Fig. 2.** Stress  $\sigma$  and temperature change ( $\Delta T_{mean}$ ) vs. strain  $\varepsilon$  obtained for Gum Metals tension up to 0.025

Increase in the specimen temperature (after it drops due to thermoelastic effect) started from point A revealed dissipative character of the process, probably caused by stress-induced phase transformation of  $\omega$  or  $\alpha''$  phases [4]. Furthermore, small increase in the temperature in this range can be a sign that the transition takes place in small volume of the alloy, what is consistent with the results of microstructure analysis [4]. From the points B and C the slope of dependence  $\Delta T$  vs.  $\varepsilon$  increases demonstrating that the deformation is irreversible from both thermodynamical and mechanical points of view. A clear tendency of the thermal plot to grow demonstrates exothermal nature of the governing deformation occurring in Gum Metal, also within the reversible strain range.

Stress strain curves derived by DIC for the Gum Metal tension at three strain rates till rupture, completed by the related temperature changes are presented in Figs 2 a, b and c. IR thermograms (left) and corresponding DIC distributions of the  $E_{yy}$  strain component (right) for the strain 0.07 (vertical dashed line in diagrams) are shown. Two kinds of the temperature of the gauge part of the specimen are analyzed: mean  $\Delta T_{mean}$ , and maximal  $\Delta T_{max}$ .

In the initial strain range the  $\Delta T_{max}$  and  $\Delta T_{mean}$  curves are overlapping; at the higher strains a gradually increasing discrepancy is observed. The point of the discrepancy (marked by a star) is an indicator of the onset of plastic strain localization. At strain rate  $10^{-3}\text{s}^{-1}$ , far from adiabatic conditions, no discrepancy is seen between the temperatures, yet the obtained non-uniform strain distribution demonstrates also the strain localization.

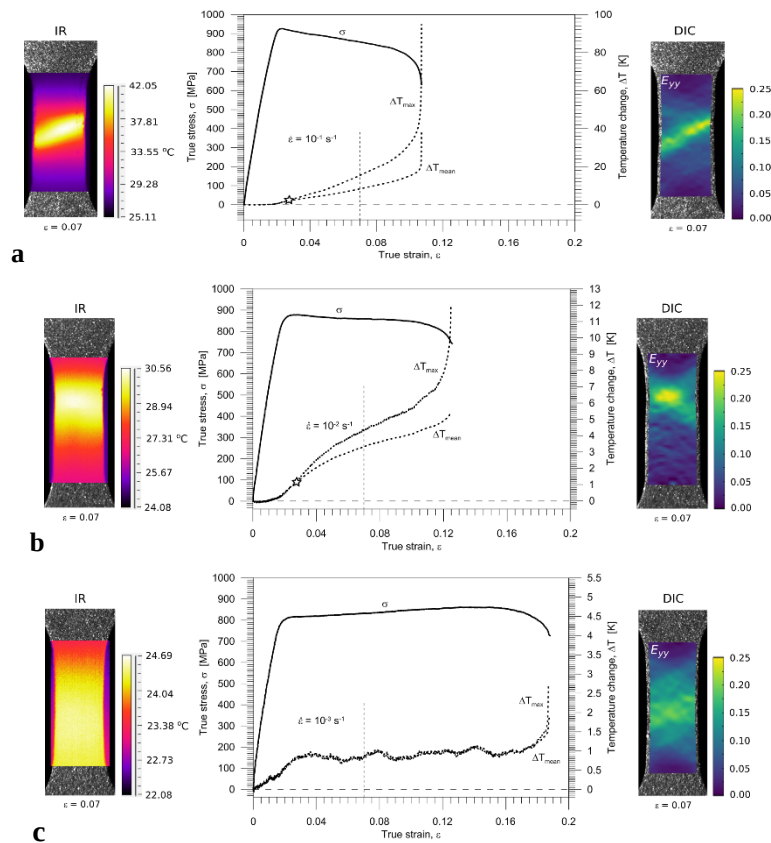


Fig. 2. Stress, maximal  $\Delta T_{max}$  and mean  $\Delta T_{mean}$  temperature vs. strain for strain rates: (a)  $10^{-3}\text{s}^{-1}$ , (b)  $10^{-2}\text{s}^{-1}$ , (c)  $10^{-1}\text{s}^{-1}$

### 3. Conclusions

Large limit of Gum Metal reversible nonlinear deformation, underlined as the unique alloy "super property", originates from mechanisms dissipative nature; exothermic stress-induced transition of  $\omega$  or  $\alpha''$  phases. During plastic deformation, both strain and temperature distributions demonstrate that at higher strain rates strain localization starts nucleating just after the yield limit.

### Acknowledgements

The research was supported by the National Science Centre, Poland; under Projects: 2014/13/B/ST8/04280 and 2016/23/N/ST8/03688.

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## TOUGHNESS TESTS OF TOOL STEELS FOR WORK AT LOW TEMPERATURE

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### 1. Introduction

On scrapyard, outside of building, scrap shear with parallel knives were used to cut large pieces of scrap metal on requested size to fill electric arc furnace. Because of several blades damages and failures, Fig 1, two different steels for blades have been analyzed to increase service life of blades and knives at low temperature conditions. Research was carried out on tool steel X45NiCrMo4 and tool steel X210Cr12 according to German standard DIN (Deutsches Institut für Normung) [1].



Fig. 1. Damaged blade

The material X45NiCrMo4 is cold work tool steel with hardness on the surface after heat treatment 53-58 HRC, with high toughness, high impact strength and compressive strength.

The material X210Cr12 is high carbon/chromium type tool steel for cold work with hardness on the surface after heat treatment 64-66 HRC, with very high wear resistances and medium toughness.

### 2. Tested material

In order to verified material, chemical composition of both steels was checked by quantometer. Chemical composition of steels X45NiCrMo4 and X210Cr12, determined by quantometer, is shown in Table 1.

Table 1. Chemical composition of steels X45NiCrMo4 and X210Cr12

| Ch.element (%)<br>Quantometer | X45NiCrMo4 | X210Cr12 |
|-------------------------------|------------|----------|
| C                             | 0.46       | 1.9      |
| Si                            | 0.15       | 0.35     |
| Mn                            | 0.38       | 0.33     |
| Cr                            | 1.22       | 12.2     |
| Mo                            | 0.22       |          |
| Ni                            | 4.04       |          |
| W                             | 0.04       |          |
| V                             |            | 0.15     |
| S                             | 0.009      | 0.005    |
| P                             | 0.018      | 0.004    |

Comparison between chemical composition of steel X45NiCrMo4 and steel X210Cr12 showed difference in three elements, table 2.

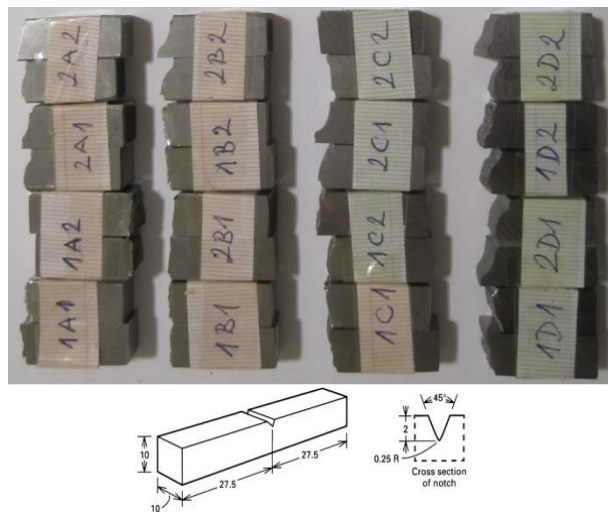
**Table 2.** Difference in chemical composition between X45NiCrMo4 and X210Cr12

| Element    | C [%] | Cr [%] | Ni [%] |
|------------|-------|--------|--------|
| X45NiCrMo4 | 0.46  | 1.22   | 4.04   |
| X210Cr12   | 1.9   | 12.2   | -      |

It is visible that steel X210Cr12 has increased content of carbon almost four times, increased content of chromium almost 10 times and no content of nickel according to steel X45NiCrMo4 which has significant content of nickel (about 4%).

### 3. Toughness and hardness tests and results

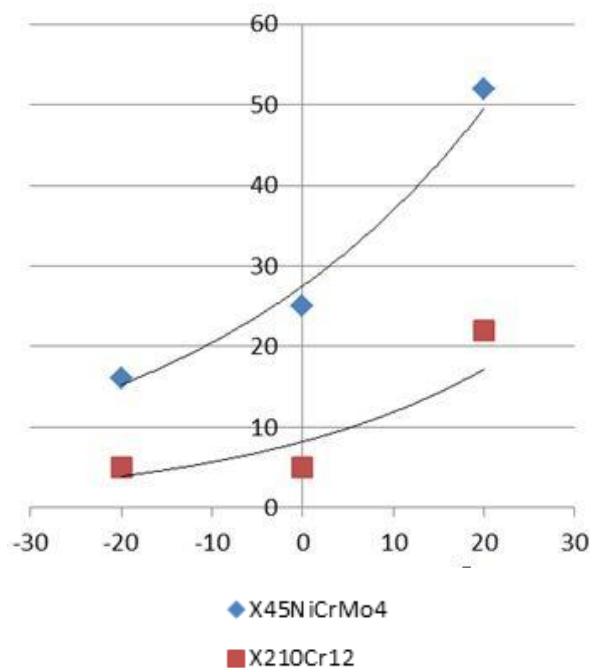
The toughness of both material, X45NiCrMo4 and X210Cr12, were determined by experimental testing in Laboratory of “Shipbuilding Industry Split”. Specimen are shown on Fig. 2.



**Fig. 2.** Shape and dimensions of the specimen

Measurement of hardness on the surface for steel X45NiCrMo4 showed 55 HRc and measurement of hardness on the surface for steel X210Cr12 showed 58 HRc.

Results Laboratory testing are shown on Fig.3.



**Fig.3.** Toughness of steels X45NiCrMo4 and X210Cr12

### 4. Conclusions

From the Fig.3 is visible that tool steel X45NiCrMo4 has better toughness on low temperatures while surface hardness of both steels is very close. Because of this results, the tool steel X45NiCrMo4 was suggested as more adequate for work during the low temperatures.

### References

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# MICRO- AND MACROSCOPIC CHARACTERIZATION OF GRANITE

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## 1. Introduction

Granites are rocks consisting of feldspar, quartz, and mica. Ever since the pioneering work of Kröner [1], scientists aim at calculating the elastic stiffness of granites based on the knowledge of the elastic properties of the three material phases. The self-consistent scheme taken from continuum micromechanics [2] is a promising candidate for establishing the described scale transition from microstructural properties of granites to their macroscopic homogenized elastic stiffness values. The self-consistent scheme accounts for key features of the microstructure, i.e. for the elastic stiffnesses of the material phases, their volume fractions, their characteristic phase shapes, and their “polycrystalline” type of mechanical interaction.

In the present work, Mauthausen granite is characterized both with microscopic and macroscopic testing techniques. The grain sizes of feldspar, quartz, and mica amount to approximately 1 mm. Microscopic characterization is carried out with the aim to provide the input values required for the aforementioned homogenization of the elastic stiffness properties of the analyzed granite. Macroscopic characterization, in turn, is carried out with the aim to provide experimental data for the quantitative assessment of the homogenization results.

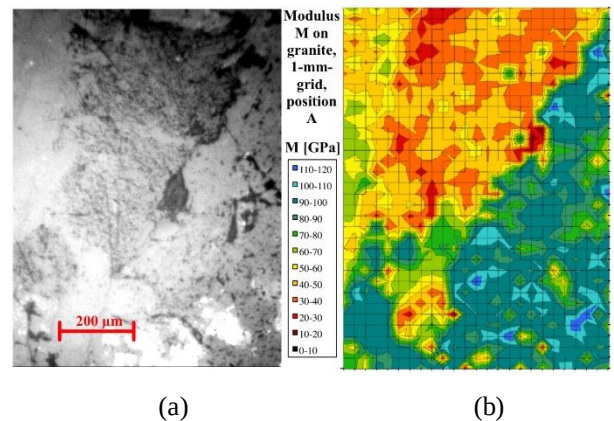
## 2. Microscopic characterization

Nanoindentation and image analysis are used to identify the elastic stiffness properties and the volume fractions of the microscopic material phases of the studied granite.

### 2.1 Nanoindentation

The first grid nanoindentation analysis is performed with the aim to identify the expected values of the indentation modulus and the indentation hardness of the material phases. Two

testing areas, referred to as A and B, are selected, see Fig. 1 for the  $0.72 \times 0.99 \text{ mm}^2$  large testing area A. Following [3,4], the two test areas are chosen in strongly structured regions of the surface. In other words, the test areas A and B show strong fluctuations of visible gray values.



**Fig. 1.** (a): Photo of testing area A. (b): Corresponding distribution of the indentation modulus.

The test grids contain  $25 \times 34 = 850$  indentation points. The grid spacing distance amounts to  $30 \mu\text{m}$ . Histograms referring to the 850 measured values of the indentation modulus and the indentation hardness are deconvoluted based on Gaussian distributions. In order to obtain residual errors which are convincingly small, it is necessary to use four rather than the expected three material phases. This is indicating that the analyzed granite contains two types of feldspars as well as one type of quartz, and one type of mica. Both testing areas A and B deliver very similar expected values of the indentation modulus and the indentation hardness of the four material phases. The indentation moduli amount to approximately 40, 60, 90, and 100 GPa, the indentation hardnesses to approximately 1.5, 3, 10.5, and 14.5 GPa.

The second grid nanoindentation analysis is performed with the aim to identify the volume fractions of the four material phases. To this end, two testing areas of  $3 \times 3 \text{ cm}^2$  are characterized

using  $20 \times 20 = 400$  indents. Thus, the grid spacing distance amounted to 1.58 mm. This is larger than the characteristic grain size. Deconvolution of the obtained histograms is again based on four Gaussian distributions, with expected values in the vicinity of the results from the first grid indentation analysis, see Table 1 for the results.

**Table 1.** Indentation moduli (M) and hardnesses (H), both in GPa, as well as phase volume fractions, resulting from deconvolution of grid indentation results

| Mica  |     | Feldspar I |     | Feldspar II |      | Quartz |      |
|-------|-----|------------|-----|-------------|------|--------|------|
| M     | H   | M          | H   | M           | H    | M      | H    |
| 39    | 1.2 | 61         | 3.5 | 88          | 10.2 | 101    | 14.3 |
| 8.2 % |     | 31.2 %     |     | 40.8 %      |      | 19.7 % |      |

## 2.2 Image analysis

The two samples used for the second grid nanoindentation analysis are scanned with a resolution of 4800 dots per inch (dpi). The resulting gray value distribution is deconvoluted based on four Gaussian distributions. The obtained phase volume fractions of the four material phases are very similar to the results obtained from the nanoindentation analysis, compare Tables 1 and 2.

**Table 2.** Phase volume fractions resulting from deconvolution of surface scans (resolution = 4800 dpi)

| Mica  | Feldspar I | Feldspar II | Quartz |
|-------|------------|-------------|--------|
| 8,1 % | 32,7 %     | 41,7 %      | 17,6 % |

## 3. Macroscopic characterization

### 3.1 Loading-unloading cycles in compression

Four granite prisms ( $16 \times 4 \times 4 \text{ cm}^3$ ) are subjected to loading-unloading cycles in compression. The deformation is measured using two independent measurement systems: strain gauges and inductive displacement transducers. Four different unloading speeds are used: 400, 800, 1600, and 3200 N/s. For each of them, ten loading-unloading cycles are performed. The obtained unloading moduli are independent of the unloading speed and of the measurement system. On average, they amount to  $48.9 \pm 1.3 \text{ GPa}$ .

### 3.2 Ultrasonic tests

The same granite samples are also analyzed using ultrasonic pulse velocity measurements with a central frequency of 2.25 MHz. Based on the theory of elastic wave propagation in isotropic media, the elastic Young's modulus is quantified based on the mass density of the granite and the determined velocities of longitudinal and transversal waves. It amounts to  $53.4 \pm 1.3 \text{ GPa}$ .

## 4. Results

The two described grid nanoindentation analyses and the image analysis delivered very similar results in terms of indentation moduli and phase volume fractions. This corroborates the credibility of the quantified microstructural properties.

As regards the macroscopic ("homogenized") Young's modulus of granite, loading-unloading in compression and the ultrasonics pulse velocity method delivered quite different results. The underlying reason is that the ultrasonics study violated the separation of scales principle: the wave lengths were on the same order of magnitude as the characteristic size of the heterogeneities, and not significantly larger [5].

## 5. Conclusions

Microstructural and macroscopic testing of granite delivered experimental data which will serve as input values for stiffness homogenization and for the quantitative assessment of the corresponding homogenization results. Anticipating that homogenization based on perfectly bonded grain boundaries will result in an overestimation of the macroscopic elastic stiffness of granite, the available knowledge regarding the macroscopic stiffnesses will allow for identifying the density of grain boundary defects.

**Acknowledgment** It is gratefully acknowledged that the analyzed granite samples were provided by Dr. A. Preh, Institute of Geotechnics, Research Center of Engineering Geology, TU Wien.

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## INFRARED THERMOGRAPHY OF STEEL STRUCTURE BY FFT

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### 1. Introduction

The Fast Fourier Transform of Infrared images is applied on real severely corroded steal structure. The method is already demonstrated on aluminum plate specimen with blind drilled holes while in presented case real steal structure is used [1]. The specimen is exploded scuba diver's tank (Fig. 1) where accident happened during the filling with fatal consequences for the operator.



Fig. 1. Severely corroded collapsed structure of scuba diver's tank.

### 2. NDT by IR thermography

The Non-Destructive Testing (NDT) by Infrared (IR) Thermography has been used for composite polymer matrix materials [2], but with the development of IR cameras, the method has found its application even for NDT of thin metal plates. Experimental setup for transient thermography is shown in Fig. 2; scuba diver's tank, IR camera – middle wave InSb focal plane array FLIR SC5000 and thermal stimuli – flash lamp.



Fig. 2. Setup for transient testing mode

Generally, metal materials are good heat conductors and it is necessary to provide a high acquisition frequency with the IR camera. Obtained acquisition frequency is 50 Hz and time of acquisition is 10 seconds per measurement giving total of 500 thermograms per measurement. In Fig. 3 raw thermal images are shown, where no traces of corroded degradations are visible.

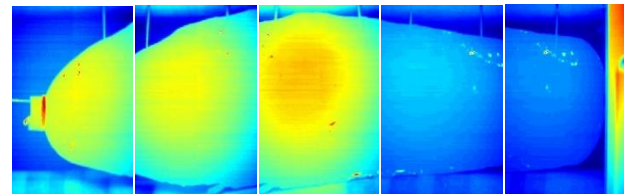


Fig. 3. Thermal images of diving

### 3. Signal reconstruction and results

After acquisition, total of 500 thermograms are post processed by Fast Fourier Transformation algorithm presented in previous work [2]. As a result, phase images are obtained. Fig. 4.a depicts phase image captured from the non damaged side (painted in black high emissivity Nextel paint, where  $\epsilon=0.975$ ). Signs of advanced corrosion can be detected in right lower corner. Deeper damages are more apparent after IR thermography testing because difference in material wall thickness is more pronounced. In damaged areas heat wave passes shorter path through the metal material resulting in different heating/cooling curves. As a result, heat waves have different phase delay and defects can be seen as different color zones on phase images. The image in Fig 4.b is a flipped photo of tank's back view, thus opposite of side on which thermograms are captured. The image is flipped to enable direct comparison of thermographic results and damages. Anomalies noticed on thermography results correspond to the area in which are located deeper rucksack corrosion areas.

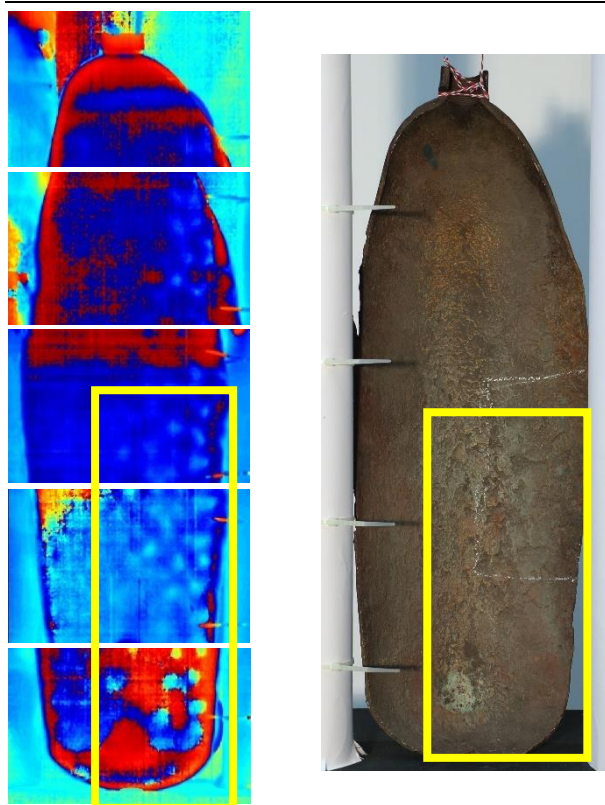


Fig. 4. a) phase image and b) photo.

#### 4. Front-side excitation results

Front-side excitation setup, shown in Fig. 5, is also considered, as NDT control should be performed as a tool for assess the severity of the damage on bottles which are still in use. Results can be seen in phase images shown in Fig. 6.



Fig. 5. Front-side excitation setup.

Authors agree that different color areas do not show damaged areas and that they are only the result of partial coverage of surface by nickel. Method should be tested on the fully nickel-free surface so that valid conclusions can be drawn.

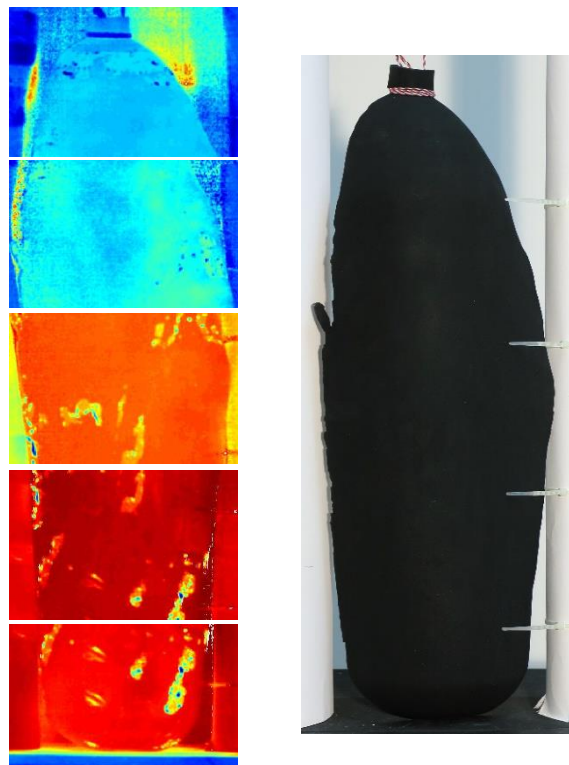


Fig. 6. a) phase image and b) photo.

#### 5. Conclusions

Proposed method can be used as powerful tool in transient thermography damage detection. Inspection of scuba diver's (or firefighting CO<sub>2</sub> tanks) is possible if straight flash bulb is produced and inserted in tank. Our current flash bulb is spiral form, that initially in production process was straight bulb. Authors agree that NDT inspection of pressure vessels is only possible in transient mode of testing due to the significant influence of surface protective layer that is not supposed to be grinded during regular inspection process.

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# MECHANICAL PROPERTIES OF COMPOSITE STRUCTURAL PARTS FOR MACHINE TOOLS

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## 1. Introduction

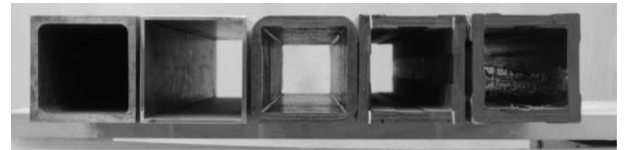
Development of new production machines is often motivated by the need to improve their mechanical properties. The most important properties are productivity, accuracy and stability. Usually this would require to create a more light-weight design to improve the machine speed and acceleration, a more rigid design to minimize the deflection of the machine under loading and also to focus on designing the structure to minimize deformations under thermal loading. The question is how the replacement of traditional isotropic materials (steel, cast iron, aluminum) with light-weight fiber composites could be beneficial, particularly in the production of machines with high demands for rigidity like machine tools.

A large overview on fiber composite application in that area was made by Möhring [1]. From the existing realizations it seems that the potential is in the application of high or ultra-high modulus carbon fibers, since these materials could fulfill the stiffness conditions. The question is how these materials could satisfy all of the demanded properties. To be able to give an answer, representative coupons of structural parts were designed and manufactured both from isotropic materials and fiber composites and their static stiffness, modal properties and thermal deformation were experimentally tested.

## 2. Coupons for Experimental Testing

Five coupons, see Fig. 1, were designed and manufactured, all with outer dimensions 130x130x1170 mm. The coupons differed in their construction material, thickness and weight. The fiber composite coupons from were made of a combination of standard high-strength carbon fibers (modulus 230 GPa) and ultra-high modulus carbon fibers (modulus 780 GPa). Two of the three composite coupons had an integrated layer of a compliant material (modulus 40 MPa) to test the

possibility of improving damping and dynamic stiffness.



**Fig. 1.** Coupons of structural parts (cast iron – steel on the left, the rest from fiber composites)

The following coupons were prepared and tested:

- Grey cast iron
  - thickness 10 mm, weight 43.4 kg
- Welded steel
  - thickness 5 mm, weight 23.2 kg
- Composite1: integral coupon made of filament winding
  - thickness 13 mm, weight 11.2 kg
  - integrated damping layers
- Composite2: coupon made of bonded plates
  - thickness 14 mm, weight 11.9 kg
- Composite3: coupon made of bonded plates
  - thickness 14 mm, weight 11.9 kg
  - integrated damping layers

The coupons were designed with an aim to keep the external dimensions identical and to achieve a similar stiffness in bending. Testing in three point bending loading, thermal loading and experimental modal analysis was performed to compare the properties of the structural parts.

## 3. Static and Dynamic Stiffness Comparison

Three point bending loading, with a 1000 mm span of the supports, see Fig. 2, was used to compare the static stiffness. Deformations and the loading forces were measured during quasi-static loading with a hydraulic cylinder. The rigidity of the coupons was evaluated from the measured dependency of the deformation on the applied force. Experimental modal analysis was performed using a modal hammer and three-axis accelerometers. To

minimize the effect of boundary conditions the coupons were hung during the measurement on a flexible rope, see Fig. 3. First mode shapes, natural frequencies and damping ratios were evaluated from the experiments. The comparison of results (stiffness, natural frequency of the first mode shapes, damping and the average damping ratio are given in Tab. 1.

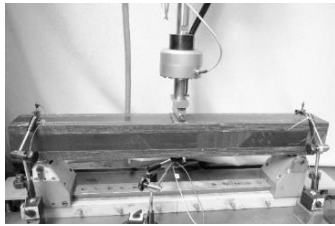


Fig. 2. Three point bending test.



Fig. 3. Experimental modal analysis

Table 1: Comparison of stiffness and modal properties

|           | Stiffness [N/mm] | $f_1$ [Hz] | $\zeta_1$ [%] | $\zeta_{avg}$ [%] |
|-----------|------------------|------------|---------------|-------------------|
| Cast iron | 73116            | 493        | 0.88          | 0.43              |
| Steel     | 74887            | 390        | 0.23          | 0.30              |
| Compo1    | 105799           | 916        | 1.54          | 1.22              |
| Compo2    | 77060            | 795        | 0.55          | 0.51              |
| Compo3    | 77046            | 762        | 1.99          | 1.22              |

#### 4. Behaviour under Thermal Loading

The configuration with one-sided clamped coupons was used for the comparison of properties under thermal loading. Thermal flow from a resistance heater was applied to the free end of the coupon. The load applied to all coupons came from the 110 V voltage in the heater, which created a 44 W heat flow. The following deformation and temperature changes were measured in several sections along the coupons, see Fig. 4. An example of the measured temperatures is given in Fig. 5. The maximum deformations and temperatures of the coupons in steady state are compared in Tab. 2.



Fig. 4. Measurement of temperatures and deformations under thermal loading.

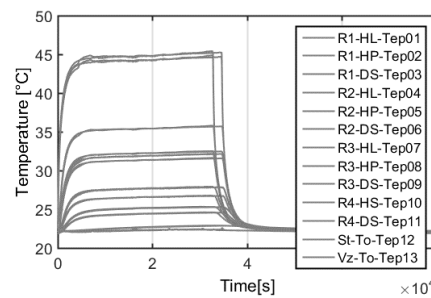


Fig. 5. Temperature behaviour under thermal loading for composite coupon.

Table 2: Properties under thermal loading

|           | Tmax [°C] | Defmax [μm] |
|-----------|-----------|-------------|
| Cast iron | 46        | 8           |
| Steel     | 56        | 10          |
| Compo1    | 41        | 17          |
| Compo2    | 50        | 49          |
| Compo3    | 49        | 48          |

#### 5. Conclusions

Composite coupons with a size of 130x13x1170 mm attained static stiffness equal or higher than that of steel or cast iron coupons with the same dimensions, while maintaining their weight 2 to 4 times lower than that of the isotropic coupons. Due to their lower weight the frequencies were significantly higher compared with those of isotropic bodies. The damping ratios of the composite parts were better than those of the cast iron body. Further improvement was possible using the damping layer without reduction of static stiffness. The smallest deformations were measured in the cast iron and steel coupons under thermal loading. The integral composite coupon showed an almost two times higher deformation although the temperature was lower than that for the isotropic coupons.

**Acknowledgements:** The results were obtained as a part of the TE01020075 Competence Center supported by the Technology Agency of the Czech Republic. The authors would also like to acknowledge the support of the company CompoTech Plus.

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# DETECTION OF SURFACE DAMAGE USING RESISTANCE TOMOGRAPHY IN THIN GRAPHENE LAYER

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## 1. Introduction

In many cases, small surface damages occur in the structures as a result of exploitation. Often, they do not cause immediate destruction, but in any case it is good to know that they occurred, and know their approximate size and location. For conductive materials, such as carbon-epoxy composites, a resistive tomography method can be used for this purpose [1, 2, 3]. Unfortunately, their relatively high conductivity makes it necessary to measure very small electrical currents, which is associated with large problems, especially the low signal-to-noise ratio. It may be a good solution to apply a graphene layer with a relatively high specific resistance to the non-conductive surface of the investigated element. Nowadays, graphene sensors are used in many measuring applications [4].

In the present study a resistance tomography method was used. The influence of the damage in the form of a hole in the conductive layer on resistance changes between electrodes in a square plate was investigated.

## 2. Experiment description

Figure 1 shows the scheme of GFRP flat specimen with thin (about 50  $\mu\text{m}$ ) graphene-polymer layer used in the present study. Specimen has the shape of 50 x 50 mm square with 18 copper electrodes on each edge. During the experiment, the resistance was measured between pairs of opposite electrodes. The measurements were made both for the sample without and with a hole. The diameter of the hole ranged between 2 mm and 5 mm. The measurements were performed with a high accuracy digital multimeter Agilent 34401A.

Numerical analysis was performed in ANSYS code using electric analysis module. Constant current flow was simulated and the voltage change was measured.

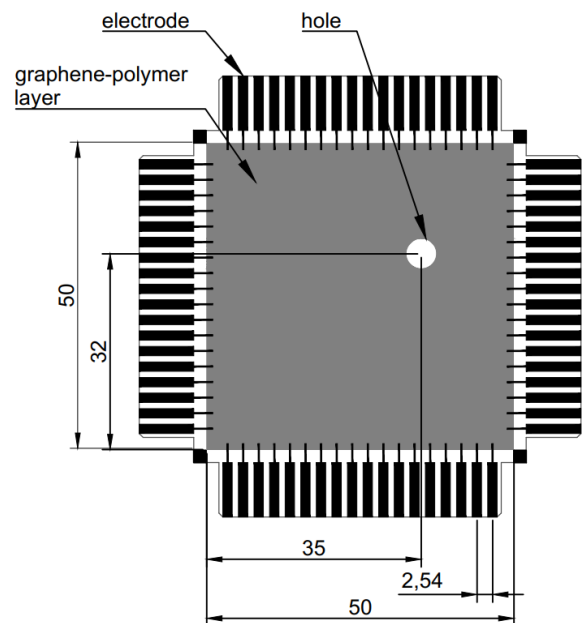
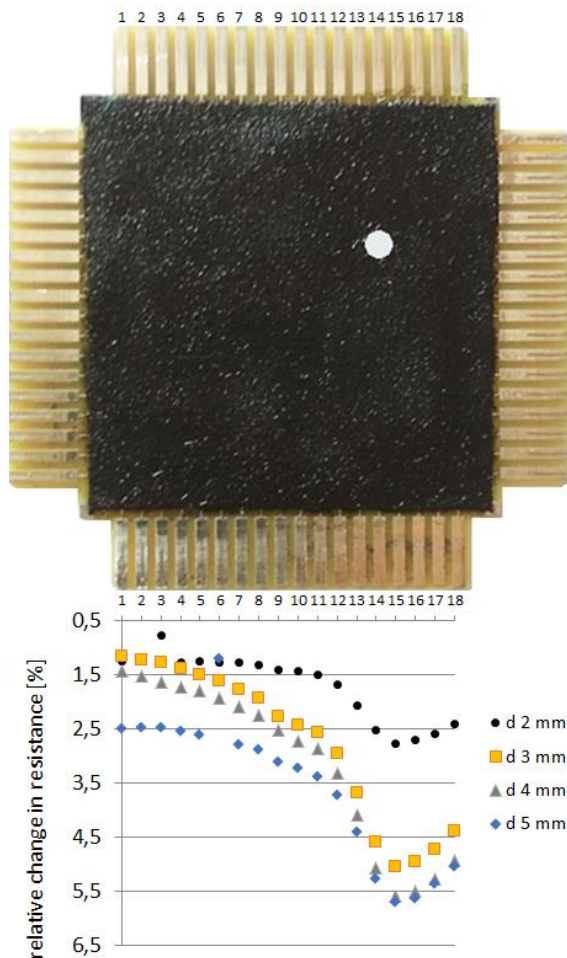


Fig. 1. GFRP plate specimen with graphene-polymer layer and cylindrical hole as a damage

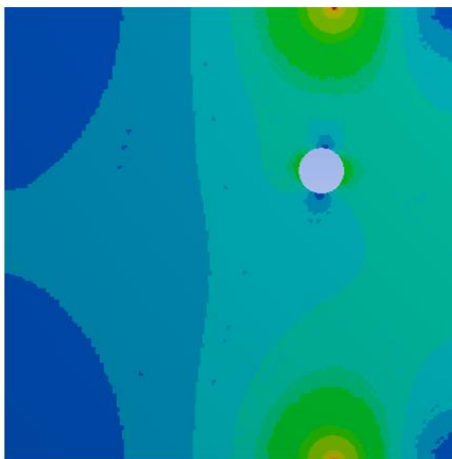
## 3. Results

Figure 2 shows the specimen and measurement results expressed by relative change in resistance for holes with different diameters. The changes are relative to the plate without damage. The resistance was measured on pairs numbered from 1 to 18. There is a visible increase in resistance caused by layer damage. This increase was observed on the entire width of the specimen and is clearly greater on the pairs between which the damage exists. The resistance change increases with increasing size of the damage.



**Fig. 2.** Experimental results: specimen and distribution of resistance difference between contact pairs from 1 to 18

Figure 3 shows an example of the electrical current density in the graphene layer. There is a visible disturbance in the current flow around the hole.



**Fig. 3.** FEM results: electrical current density for the couple of electrodes number 14

## 4. Conclusions

In both the FEM analysis and experiment, a relationship between distribution of the resistance change and the size of the damage was observed.

The proposed method can be used for detection of damages on the surface of elements and for estimation of their size and location.

## Acknowledgements

Authors would like to thank QENERGY Ltd. for providing graphene nanoplates for research.

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# WATER PERMEABILITY OF HOT MIX ASPHALT

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## 1. Introduction and Objectives

Hot Mix Asphalt (HMA) mixtures as the main construction material for road pavements are a composite of bitumen as the binder and mineral aggregates with a predefined mix design. Asphalt pavement structures consist of several layers, each with certain tasks.

Surface layers, in particular are responsible to distribute traffic loads, to ensure road safety by having high skid resistances and to seal the structure from water. Stiffness, quality of the aggregates and the mix design play an important role. Thus, one key element is the bulk density of the compacted layer. Bulk densities or rather air void contents (AVC) have an influence on the mechanical behavior and serviceability of hot mix asphalt layers. Therefore, conventional design approaches set maxima and minima for air void contents of hot mix asphalt pavements and use it as a sign of quality for the construction process to comply with the criteria of the road administration. Minimum limits for AVC are necessary to prevent the pavements from bleeding effects of the bitumen. Bitumen films due to bleeding will lower the friction on pavement surfaces. Maximum limits for AVC are set to ensure a sufficient layer stiffness that is necessary to distribute stresses from mechanical impacts [1] and to prevent water from entering the pavement structure. Based on experience, AVC limits are given for commonly used pavement designs [2]. When developing new mix designs using innovative and alternative materials, initial proof of concept studies have to be carried out, since no empirical data are available to ensure that all requirements are fulfilled. Regarding the function as a sealant layer, a water permeability test is helpful. In the U.S., different methods for analyzing the water permeability in the laboratory [3] or in the field [4] were developed.

In this paper, a method to measure the water permeability is presented using HMA specimen sizes that are commonly used in Austria. With this

procedure, coefficient of permeability (COP)  $k$  can be determined to be used for further calculations (e.g. flow calculations for asphalt sealing elements in hydraulic engineering). Water permeability tests for soils are already standardized in German standards [5] with defined permeability classifications using coefficient of permeability  $k$ . These classifications are applied on bituminous pavements. Table 1 contains the classes given by [5].

**Table 1.** Scale of permeability [5]

| $k$ [m/s].             | scale               |
|------------------------|---------------------|
| Below $10^{-8}$        | Very low permeable  |
| $10^{-8}$ to $10^{-6}$ | low permeable       |
| $10^{-6}$ to $10^{-4}$ | permeable           |
| $10^{-4}$ to $10^{-2}$ | very permeable      |
| above $10^{-2}$        | Extremely permeable |

## 2. Materials and Methods

An asphalt concrete with a maximum nominal aggregate size of 11mm (AC 11) was employed. The binder content was set to 5.2% by mass. Five HMA Slabs (50x26x4 cm) were compacted in a roller compactor. From the slabs, 40 specimens (8 each) are cored with a diameter of 100mm and a height of 40mm. The air void content of the specimens range from 4.0 to 9.0 vol%.

Water permeability tests are carried out on all specimens. The method, that was used as described below, is based on the water permeability test for soils, standardized in German Standards [5]. The setup and equipment is shown in figure 1. HMA specimens are located within a triaxial cell between filter stones and are covered by an elastic membrane. The triaxial cell is flooded with ambient air. To enable the water flow through the specimens, a confining pressure has to build up in lateral direction. Thus, an overpressure of 150 kPa was applied within the triaxial cell. Due to the higher lateral pressure, the elastic membrane is pressed onto the specimen surface and the water is forced to flow through the specimen, instead of passing on the outside. The tests were carried out three times on

each specimen. Starting with a flow pressure of 50 kPa, it was increased to 70 and 100 kPa for the second and third test. The specimens are flown through with water until the system is saturated. To measure the used flow pressure accurately, a pressure gauge was installed directly before the inlet to the triaxial cell. Once the test is started, water will exit the system into a measure vessel. With recording time, flow pressure and water amount, it is possible to calculate the coefficient of permeability  $k$ , which is shown in equation 1.

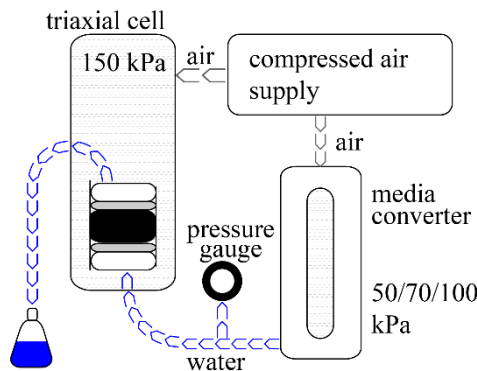


Fig. 1. Water permeability set-up

$$k = Q * \frac{L}{h} * \frac{1}{A * t} \quad (1)$$

$k$ ... coefficient of permeability [m/s]

$Q$ ...flow volume [m<sup>3</sup>]

$L$ ... specimen height[m]

$A$ ... specimen cross section[m<sup>2</sup>]

$t$ ... testing time [s]

$h$ ... pressure head [m]

### 3. Results and Outlook

Main results are shown in figure 2. The diagram shows AVC vs. the coefficient of permeability  $k$  for all 40 specimens. The mean values of three tests, varying the flow pressure were used. Furthermore, the scales of permeability according to [5] are stated. The correlation of both values, AVC and COP  $k$  is shown by applying a power function. The coefficient of determination  $R^2$  is 0.7, which indicates a fair correlation. Looking at the scales of permeability, it can be seen, that almost every data point can be classified in this areas. Therefore, according to [5], specimens with AVC lower than 5.0 vol% can be classified as “very low permeable”, between 5.0 vol% and 7.1 vol% as “low permeable” and above of 7.1 vol% as “permeable”. Comparing that with classifications according to [2] that are based on experience in Austria, the results can be

compared. Pavements are as defined as “permeable” with AVC higher than 8.0 vol%. Between 5.0 and 8.0 vol%. it is classified as “low permeable” and in the range from 3.0 to 5.0 vol% “practically impermeable”. It can be stated that no significant influence of varying the flow pressure (50, 70, 100 kPa), can be seen in the results.

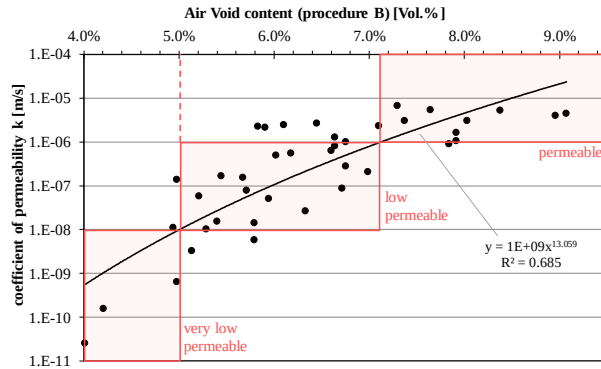


Fig. 2. Air Void Content vs. coefficient of permeability  $k$  for water

The presented method to measure the water permeability of HMA specimens is suitable to achieve coefficient of permeability  $k$  for air void ranges of approximately from 3.5 to 10.0 vol% . The correlation of the presented data in the range from 5.0 to 7.5 vol% can be improved by using higher specimens heights. Using slim specimens, air voids can be distributed inhomogeneously and can thus act as drainages.

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## ON THE COMPRESSIBILITY OF RUBBER: EXPERIMENTS AND THEORETICAL CONSIDERATIONS

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### 1. Introduction

The extrusion process typically comprises the heating of the material (polymers, rubber compounds, rubber blends), the pressing or squeezing of this material through formative tools (exposing it to high pressure of up to 100 MPa), and, upon leaving these tools, a hardly predictable shape change, also known as die swell [1]. The die swell effect is the reason why the design process of the formative tools is – so far – based on a time-consuming and ineffective trial-and-error procedure. Accurate computational simulation of the flow behavior of (unvulcanized) rubber is believed to significantly improve the efficiency in the manufacturing of the dies, which are eventually applied in rubber extrusion. However, despite considerable ongoing and previous efforts, this task remains an open challenge. Evidently, one of the reasons for this unsatisfactory situation is insufficient consideration of, or knowledge on, the mechanical compressibility of the extrudate material [2]. The present experimental campaign aims at closing some of these gaps by specifying the actual type of compressibility, studying its elastic or viscous nature, and testing its pressure sensitivity.

### 2. Materials and methodology

In this work, three different types of rubbers, namely one natural rubber (NR) and two EPDM (ethylene-propylene-diene-monomer) rubber compounds, were tested. The two EPDM rubber compounds differ from each other in terms of the type of polymer (crystalline or amorphous) and the filler content.

In order to study the mechanical compressibility of the material, a series of hydrostatic compression tests was carried out by means of a capillary rheometer, see Fig. 1. The tests followed a well-

defined sequence of steps including (i) filling of the rheometer's extrusion canal, (ii) extrusion of excessive material in order to receive the desired material specimen, (iii) closing of the canal, and (iv) hydrostatic loading and unloading. During stage (iv) the resulting hydrostatic pressures and volume/length changes were recorded, see Fig. 2. In order to consider possible dependencies on loading speed, maximum pressure and overall loading time, those factors were varied in the course of the testing campaign.

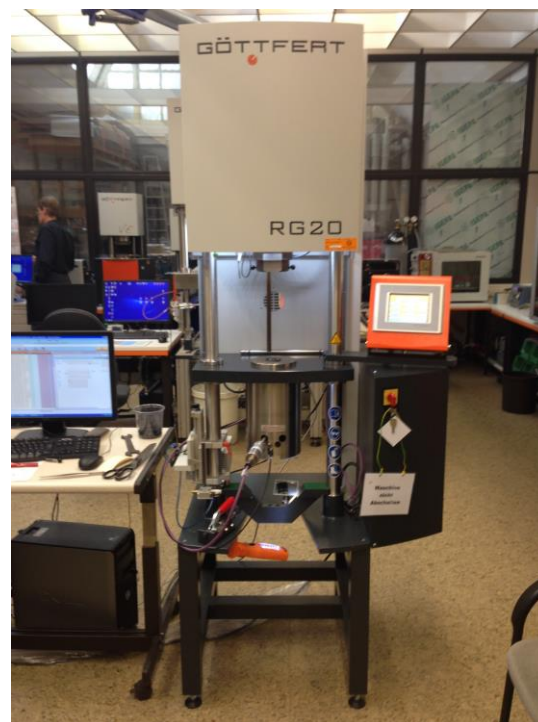
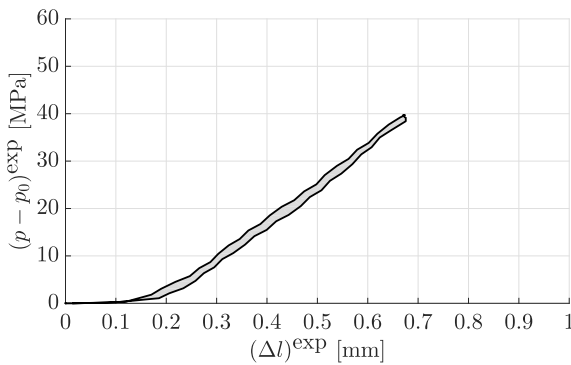


Fig. 1. Frontal view of capillary rheometer

### 3. Results

The recorded volume changes entered various relevant strain measures (Green-Lagrange strains  $E^{GL}$ , Hencky strains  $E^H$ , linearized strains  $\epsilon$ ), while the pressures were mathematically transformed into

energetically conjugate stress measures (second Piola-Kirchhoff stress  $\pi$ , Kirchhoff stress  $\tau$ , Cauchy stress  $\sigma$ ). Insertion of these measures into the dissipation inequality resulting from the two fundamental laws of thermodynamics revealed that the investigated materials behave purely elastically under hydrostatic pressure, albeit in a non-linear fashion, again see Fig. 2.



**Fig. 2.** Experimental data recorded during a single test carried out on material NR, presented in the format of a loading-unloading curve; the shaded area represents eventually dissipated energy

It could also be shown that the elastic bulk modulus  $K$  of all materials increase non-linearly with the hydrostatic pressure  $p$ , with the increase being degressive, see Fig. 3. Thereby, the NR material is softer than the two EPDM compounds, while the considered stress and strain measures do only cause slight differences in the results. Other parameter variations (loading speed, maximum pressure, etc.) do not influence the resulting bulk moduli. However, the bulk modulus relates to the hydrostatic pressure via a power function of the form given by

$$\frac{K}{K_{\text{ref}}} = \left[ \frac{p - p_0}{K_{\text{ref}}} \right]^{\beta}, \quad (1)$$

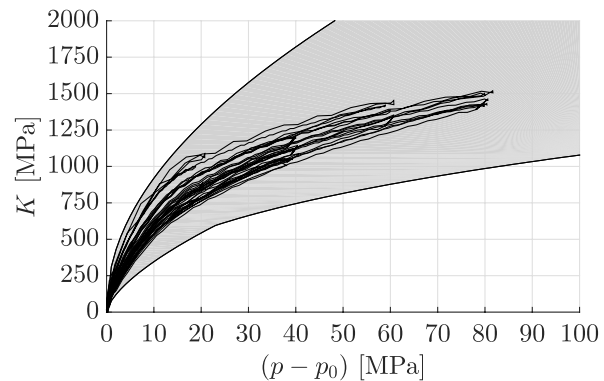
where  $K_{\text{ref}}$  and  $\beta$  are fitting parameters and  $p_0$  is an initial pressure. Further variation of testing parameters also revealed that the bulk modulus depends on the temperature, with the former decreasing when the latter is rising.

Statistical evaluation of the fitting parameters of the power function allowed for derivation of upper and lower bounds of the bulk modulus as functions of the hydrostatic pressure (and of the temperature), for details see [3] and Fig. 3.

#### 4. Conclusions

The present experimental campaign revealed remarkable compressibility of natural and EPDM

rubber and allows for mathematical modelling by use of a power function. Additionally, the purely elastic behavior under hydrostatic compression could be shown. In the future, the new constitutive relations revealed in this work allow, on the one hand, for improving the accuracy of rubber extrusion simulations [4], and, on the other hand, for optimized material characterization in terms of the number of required tests per material.



**Fig. 3.** Development of the bulk modulus  $K$  over pressure  $p$  for material NR, evaluated in terms of linearized strains and Cauchy stress, including 90%-confidence regions of the bulk modulus

#### Acknowledgements

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# EFFECT OF AS-CAST SURFACE LAYER ON ALUMINIUM ALLOY FATIGUE STRENGTH

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## 1. Introduction

The usage of cast aluminium components contributes to continuously increasing lightweight demands in mechanical engineering. However, sand cast components possess both volumetric and as-cast surface based heterogeneities. These manufacturing dependent material features affect the fatigue strength of such components [1]. While the detrimental effect of spatial heterogeneities such as shrinkage or gas pores is the topic of several recent investigations, the effect of surface layer integrity is often studied only for machined conditions [2]. The improved assessment of surface layer effects, especially the combination of microstructural imperfections and surface based micro-notches are technologically utmost relevant and define the aim of this research work.

### 1.1 Effect on bulk fatigue strength

Volumetric fatigue strength is commonly based on the largest defect size. Murakami's  $\sqrt{\text{area}}$  approach features an industrial applicable design tool, which is based on the projected cross-section of the defect perpendicular to the principal stress direction as fatigue strength qualifier [3]. To determine the extremal-sized defect acting as crack initiator, the statistical distribution of imperfections needs to be incorporated.

### 1.2 Influence of surface topography

As-cast surfaces are defined by its roughness parameters in EN ISO 4287, such as arithmetical mean  $R_a$ , total height  $R_t$  and mean roughness depth  $R_z$ . To consider the fatigue strength reducing effect, standards such as the FKM-guideline utilize those roughness parameters to calculate simplified surface effect design factors. However, the manufacturing process dependent three-dimensional surface topography is in general not evaluated in detail within such recommendations.

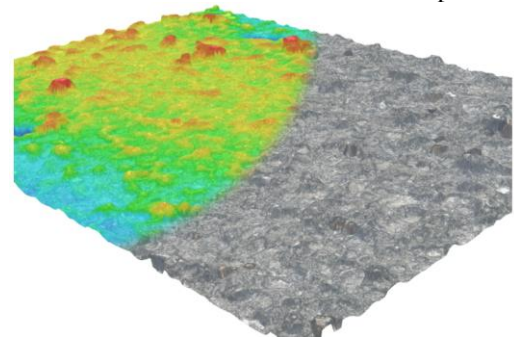
Arola determines the notch effect not only by the mean roughness depth value, but by several

roughness parameters [4], see Eq. (1). It should be mentioned that the author investigated only machined surfaces. In his approach, the radius  $\rho$  of the surface topography based micro-notch as well as a load case dependent factor  $n$  are considered.

$$K_t = 1 + n \cdot \left(\frac{R_a}{\rho}\right) \cdot \left(\frac{R_t}{R_z}\right) \quad (1)$$

## 2. Surface roughness evaluation

To characterize the surface profile of each specimen, the three-dimensional topography is firstly scanned with a digital light microscope. An example of a scanned as-cast surface is shown in Fig. 1. Utilizing the generated 3D data point cloud, the roughness parameters are calculated by a user-defined routine. Both a line-based profile evaluation as well as an area-based calculation is implemented.



**Fig. 1.** Digital topography scan of cast Al-alloy surface (height colour map versus actual profile)

By using a line-based roughness assessment, some of the critical surface characteristics may not be detected. It was found that about a dozen, randomly positioned, line measurements provide similar parameters as the area-based calculation in terms of arithmetical mean area parameter  $S_a$  and total height area profile value  $S_t$ .

Gaining information about the local  $S_a$  and  $S_t$  values is accomplished by plotting the evaluated area-based roughness distribution over the image of the specimen. After fatigue testing, the fracture information about the technical crack initiation point is additionally marked as red dot, see Fig. 2.

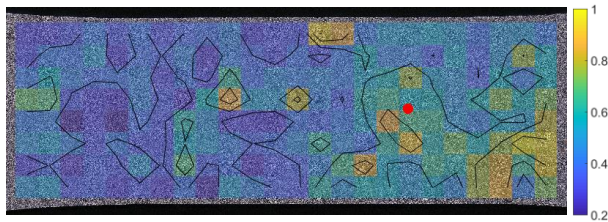


Fig. 2. Area-based Sa roughness plot (normalized)

### 3. Experimental work

Fatigue strength results of flat specimen bending tests possessing as-cast surface are shown in Fig. 3. It is clearly visible that no distinct endurance limit exists. This is due to crack initiation at shrinkage porosity and surface micro notches. The scatter index is comparably huge and implies quite severe differences in surface topography as well as surface layer porosity.

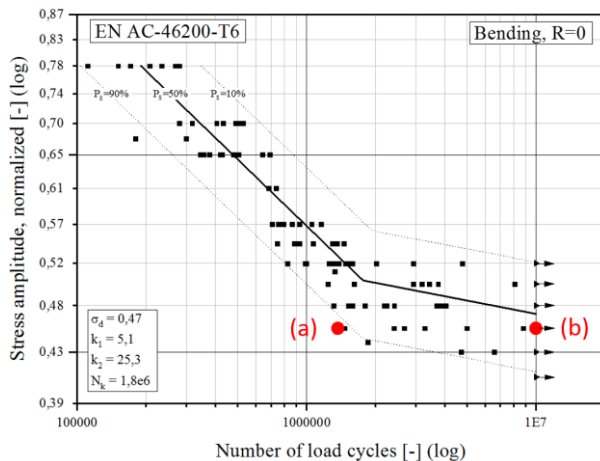


Fig. 3. Normalized S/N curve

The differences in fatigue life are investigated further for two specimens at a load level close to the endurance limit. These samples are marked with two red dots, labelled as (a) and (b) in Fig. 3.

#### 3.1 Fracture surface analysis

To evaluate both the micro-notch at the surface and the near-surface located porosity, each fracture surface is investigated by scanning-electron-microscopy (SEM). Two characteristic fracture surface images are shown in Fig. 4.

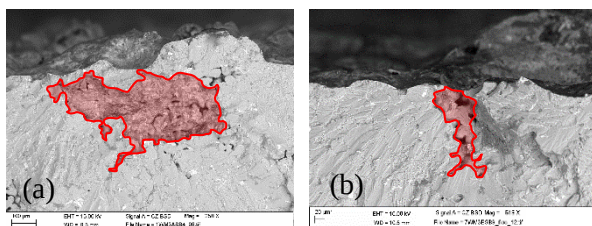


Fig. 4. SEM images of two characteristic defects

The equivalent circle diameter ranges from 430  $\mu\text{m}$  (a) to 100  $\mu\text{m}$  (b) and, the surface micro-notch radius is evaluated as 450  $\mu\text{m}$  (a) to 30  $\mu\text{m}$  (b). It was found that the surface notch radius should be at least the total height area profile. The measured effective defect area and the derived surface stress concentration factors are now applicable to estimate fatigue strength tendencies.

#### 3.2 Discussion

In case that the micro-pore broaches the surface topography, as in the currently evaluated two specimen showcase, both Arola's surface stress concentration factors (9% difference) and Murakami's fatigue strength ratio (27%) have to be combined. This leads to an endurance limit ratio of about one to 1.16, which basically matches the examined test results.

### 4. Conclusion

To assess the surface integrity of as-cast aluminium alloys, the surface-based stress concentration needs to be superimposed with the decreased fatigue strength due to micropores, as exemplified for the given showcase. Future work focusses on the statistical fracture surface and roughness profile evaluation to obtain a holistic surface layer fatigue strength assessment.

### Acknowledgements

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## EARLY-AGE CHARACTERIZATION AND MULTI-SCALE MODELING OF POLYMER-MODIFIED CEMENT PASTES

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### 1. Introduction

The early-age evolutions of the elastic stiffness and of the creep properties of polymer-modified cement pastes are characterized based on macroscopic hourly-repeated three-minutes-long creep tests [1,2]. The corresponding reaction kinetics of the materials are quantified using quasi-isothermal differential calorimetry. Combination of the described experimental methods results in known evolutions of the elastic Young's modulus and the creep modulus, as functions of the degree of hydration of the polymer-modified cement pastes. These experimental data serve as input for multiscale analysis [3,4]. In the framework of a top-down identification approach, the creep behavior of the microscopic polymer particles is identified.

### 2. Macroscopic early-age characterization

The experimental protocol of Irfan-ul-Hassan et al. [1] is used for early-age characterization of polymer-modified cement pastes. The tested materials are stored and tested 20°C. Cylindrical specimens with a height of 30 cm and a diameter of 7 cm are demolded 20 hours after production. The first three-minutes-long creep test is carried out as soon as 21 hours after production.

The test setup includes bottle-necked cylinders which ensure a close-to-perfect central load application [1]. Thus, the creep tests are carried out under a compressive normal force, while bending of the specimens is limited to the realizable minimum. The forces are measured with the load measurement cell of the testing machine. The deformation of the specimens is measured in their central regions, using five inductive displacement transducers. They are clamped to the specimens using two aluminum rings. The measurement length amounts to 164 mm.

The intensity of the external loading is small enough such that the specimens remain undamaged and exhibit a linear creep behavior [2]. At the same time, the loading is large enough as to ensure that the readings of the inductive displacement transducers are reliable, see [2] for details.

Three-minutes-long creep tests are performed once every hour. Three minutes are short enough as to ensure that the chemical hardening reaction between cement and water does not progress significantly. Thus, every test characterizes one specific *non-aging* microstructure. The 57 minutes lying between two successive creep tests, in turn, are long enough as to ensure that creep recovery is completed, i.e. there is no influence of one creep test on the following creep tests. This way, in total 168 three-minutes-long creep tests are performed between 21 hours after production and a material age amount to 8 days.

Test evaluation rigorously distinguishes between time-independent elastic deformation and time-dependent creep deformation [1,2]. Even during fast loading and unloading, it is considered that the measured deformation contains both elastic and creep deformation. Using a power-law-type creep function, test evaluation delivers values of the elastic Young's modulus and of the creep modulus of macroscopic polymer-modified cement pastes.

The corresponding reaction kinetics of the materials are quantified using quasi-isothermal differential calorimetry. Combining the temporal evolutions of the mechanical properties with the temporal evolution of the degree of hydration allows for determination of mechanical properties as a function of the degree of hydration.

### 3. Multiscale modeling

Multiscale modeling is carried out in the framework of continuum micromechanics. It follows the lines of Königsberger et al. [3]. A two-scale representation is used for polymer-modified cement pastes [4]. Cement paste is considered to be a matrix-inclusion composite consisting of a hydrate foam matrix and cement grain inclusions. At that scale of observation, the hydrate foam appears to be homogeneous. At the next smaller scale of observation, in turn, its heterogeneous microstructure becomes visible. The hydrate foam is a polycrystalline arrangement of needle-shaped hydration products (which are isotropically oriented in all space directions), spherical capillary pores, and spherical polymer particles.

Homogenization of non-aging viscoelastic properties is based on the correspondence principle [3,4]. The time-dependent problem is transformed into the Laplace-Carsen space. There, the problem manifests itself as a sequence of quasi-elastic homogenization tasks. This is solved based on the self-consistent scheme for polycrystalline microstructures and on the Mori-Tanaka scheme for matrix-inclusion composites. The obtained solution is finally back-transformed from the Laplace-Carsen space to the regular time domain.

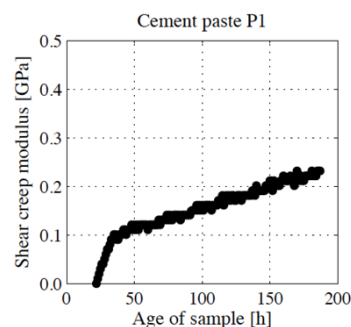
Multiscale modeling aims at resolving material microstructures down to scales of observation at which material *constants* (rather than evolving material *parameters*) can be assigned to the elementary material constituents. Based on the described two-scale representation, this approach was successful for classical cement pastes. In [3], an isochoric creep tensor function was introduced for hydration products and the related viscoelastic constants were identified. As for polymer-modified cement pastes, also an isochoric creep tensor function is introduced [4]. Still, it turns out to be impossible to assign creep *constants* to polymers.

In order to achieve a satisfactory agreement between the measurements recorded during macroscopic creep testing and the corresponding homogenization results, it is necessary to consider that the shear creep modulus of polymers is an evolving material property, see Fig. 1.

### 4. Conclusions

The creep shear modulus of polymers evolves bilinearly, see Fig. 1. During mixing, it is likely that polymers form agglomerates. During hardening of cement pastes, needle-shaped hydrated grow into

these agglomerates, such that they are split up into individual polymer particles. This explains the first steep increase of the shear creep modulus. Subsequently, the self-desiccation-related reduction of the internal relative humidity is likely to reduce the water content inside the polymer particle, such that they stiffen progressively. This explains the more moderate increase of the shear creep modulus.



**Fig. 1.** Evolution of the shear creep modulus of microscopic polymer particles, identified from a sequence of 168 three-minutes-long creep tests on polymer-modified cement paste P1, see [4] for details.

### Acknowledgements

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# MICROMECHANICAL ANALYSIS OF FIBER SPATIAL DISTRIBUTION INFLUENCE IN UNIDIRECTIONAL COMPOSITE CROSS-SECTION ON OVERALL RESPONSE IN TERMS OF TENSILE CURVES

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## 1. Introduction

Composites are widely and commonly used in industry at present. One important part of a design process of the composites is investigation of quality of such construction material. There are number of characteristics which could influence the quality of final composite material, first of all fiber volume fraction, the next are for example UV stability and ability not to absorb air moisture. Fiber volume fraction directly influences mechanical properties of resulting composite. Nevertheless it is not the only quality which can directly influence mechanical behavior of the whole composite.

Presented analyses are focused on unidirectional composite made of epoxy matrix reinforced by carbon fibers. Mechanical behavior of the whole composite does not depend on fiber volume fraction only. It depends even on spatial distribution of the fibers especially throughout the cross-section perpendicular to the fibers.

This study is a continuation of previous work which focused on the influence of the fiber spatial distribution on resulting elastic parameters of unidirectional lamina. This study uses the same description of the irregularity of the fiber spatial distribution ( $\hat{\Gamma}$  – degree of irregularity), but focuses on the influence of this parameter on stress-strain dependencies obtained in different directions against applied loading force. Non-linear anisotropic elastic material model for fibers and elastic-plastic isotropic model for matrix are considered for finite element analyses.

## 2. Experiment

Stress-strain dependencies until rupture were measured in seven loading directions.

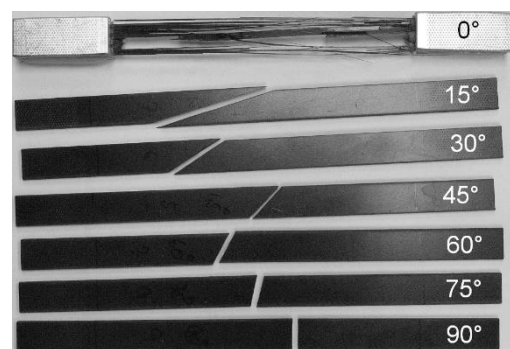


Fig. 1. Measured specimens.

## 3. Material models and parameters

Non-linear transversely isotropic elastic material model for fibers was used. The only nonlinearity was considered in fibers' axis direction. Young's modulus in fibers' axis direction  $E_{11}$  depends on strain  $\varepsilon_{11}$  in the same direction

$$E_{11}(\varepsilon_{11}) = E_{11}^0 (1 + g \varepsilon_{11}), \quad (1)$$

where  $E_{11}^0$  is initial Young's modulus and  $g$  is nonlinear parameter.

Elastic-plastic isotropic model with Von-Mises plasticity is used for matrix. Hardening function is considered in the form

$$\sigma_y = \sigma_y^0 + \frac{P_E \bar{\varepsilon}_P}{\left(1 + \left(\frac{P_E \bar{\varepsilon}_P}{P_A + P_K \bar{\varepsilon}_P}\right)^n\right)^{1/n}}, \quad (2)$$

where  $\sigma_y$  is current yield stress,  $\sigma_y^0$  is initial yield stress and  $P_E$ ,  $P_K$ ,  $P_A$  and  $n$  are shape parameters.

## 4. Identification of material parameters

First, the identification of the material parameters of fibers and matrix was performed using unit cell element with regular fiber distribution (shown below in figure 2). Identification was performed using gradient based

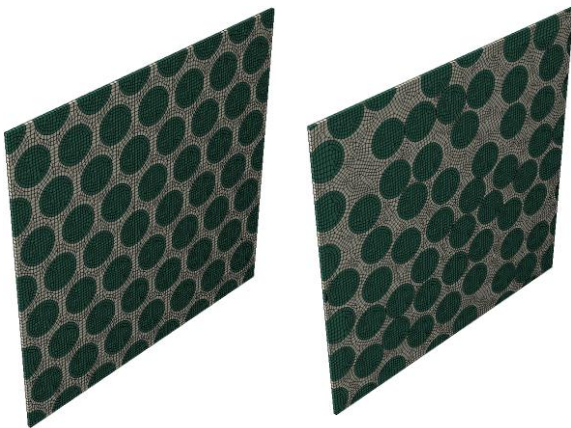
optimization method. Identified material parameters are shown in table 1.

**Table 1.** Identified material parameters of fiber (top part) and matrix (bottom part).

|                                               |       |        |
|-----------------------------------------------|-------|--------|
| $E_{11}^0$                                    | [GPa] | 179.47 |
| $E_{22}^f = E_{33}^f$                         | [GPa] | 16.88  |
| $\nu_{12}^f = \nu_{13}^f$                     | [-]   | 0.25   |
| $\nu_{23}^f$                                  | [-]   | 0.40   |
| $G_{12}^f = G_{13}^f$                         | [GPa] | 55.44  |
| $G_{23}^f = \frac{E_{22}^f}{2(1+\nu_{23}^f)}$ | [GPa] | 6.03   |
| $g$                                           | [-]   | 9.93   |
| $E_m$                                         | [GPa] | 3.3    |
| $\nu_m$                                       | [-]   | 0.33   |
| $\sigma_y$                                    | [MPa] | 23.14  |
| $P_E$                                         | [GPa] | 9.82   |
| $P_A$                                         | [MPa] | 119.82 |
| $P_K$                                         | [MPa] | 764.49 |
| $n$                                           | [-]   | 0.86   |

## 5. Finite element micromodel

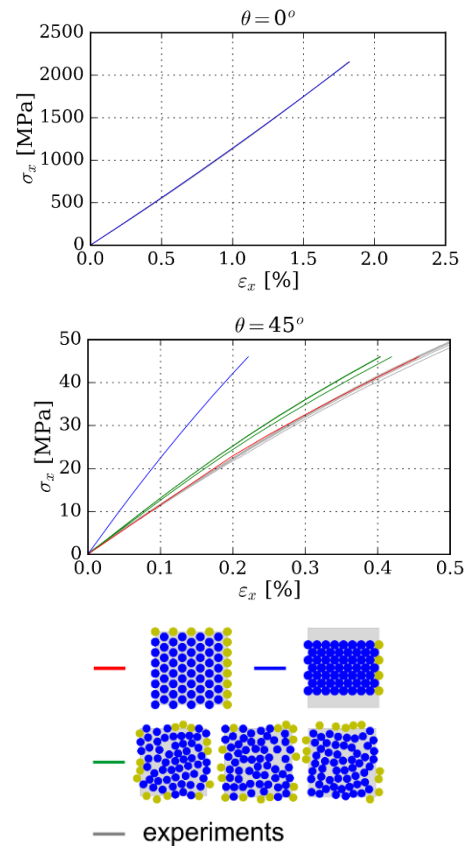
Finite element analyses were performed in Abaqus software. Subroutine written in Fortran was used for definition of constitutive relation for fibers. The whole process of the finite element model building (definition of periodic boundary conditions, automatic submitting of analyses, processing and printing of results) was controlled by Python scripts.



**Fig. 2.** Regular (left) and example of automatically generated random finite element mesh unit cell (right).

## 6. Influence of fiber spatial distribution

Stress-strain dependencies for cases when loading force form 0° and 45° angle with fiber direction. Experiments and unit cells with 5 different spatial distribution of fibers are shown.



**Fig. 3.** Stress-strain dependencies for chosen unit cells with different spatial distribution of fibers.

## 7. Conclusions

Spatial distribution of fibers in cross-section perpendicular to the fibers has significant influence on the resulting mechanical properties of the lamina.

## Acknowledgements

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# VALIDATION OF IDENTIFIED MATERIAL PARAMETERS OF RUBBER USING AN ARCAN SHEAR TEST

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## 1. Introduction

More than one type of test is crucial in order to identify robust material parameters of rubber. Experimental data can be obtained from tests in uniaxial tension, uniaxial compression, equibiaxial tension, pure shear, simple shear, etc [1].

Uniaxial tension and uniaxial compression tests were performed to obtain experimental data for identification of material parameters in this work. The Mooney-Rivlin hyperelastic material model was used in order to characterize the rubber investigated.

Because of the variability of the test configuration (possibility to change the tensile and shear deformation ratio of the Arcan sample), the Arcan shear test [1] seems to be a suitable experiment for the validation of the material model. The aim of this paper is to demonstrate the influence of various types of clamping used in the Arcan shear test on the comparison of the test and its numerical simulation.

## 2. Identification of material parameters

The force-displacement curves were obtained using the uniaxial tension and uniaxial compression tests (Fig. 5) in the Zwick/Roell Z050 universal testing machine. The temperature was  $23 \pm 1$  °C; the atmospheric moisture was  $50 \pm 6\%$ . A mechanical extensometer was used to measure displacements. Four load-unload cycles were performed for each of the deformation levels, which reached the strain of 5%, 15%, and 25%. The strain rate of  $1.25 \cdot 10^{-2} \text{ s}^{-1}$  and  $4.2 \cdot 10^{-3} \text{ s}^{-1}$  was prescribed in the first three cycles and in the fourth cycle of each deformation level, respectively.

The Mooney-Rivlin model for incompressible hyperelastic materials was used. The strain energy density function  $W$  is in this case

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3), \quad (1)$$

where  $C_{10}$  and  $C_{01}$  are the material parameters and  $I_1$  and  $I_2$  are the first and the second invariant of the right Cauchy-Green deformation tensor [3].

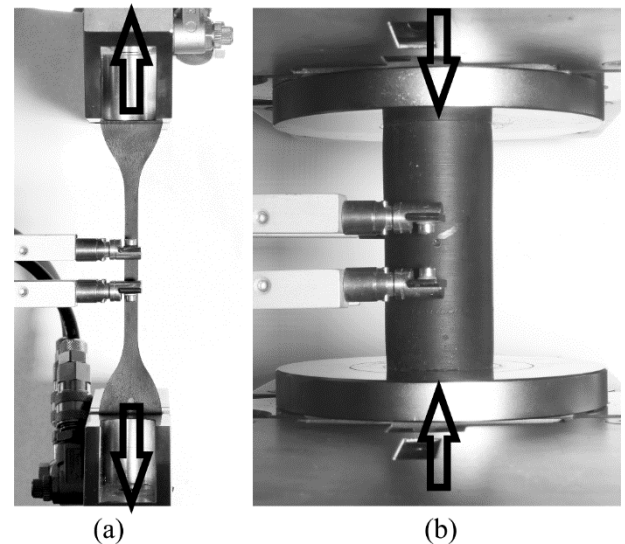


Fig. 5. a) Uniaxial tension test, b) uniaxial compression test.

The material parameters were identified using the optimization algorithms in SciPy software [4] based on the stress-strain curves. In the measured volume of tension and compression samples, homogeneous deformations were assumed.

## 3. Arcan shear test

The Arcan shear test was also performed using the Zwick/Roell Z050 universal testing machine. Because the special Arcan clamps can be rotated in the testing machine clamps (see Fig. 6), tensile and shear deformation can be combined in a selected ratio.

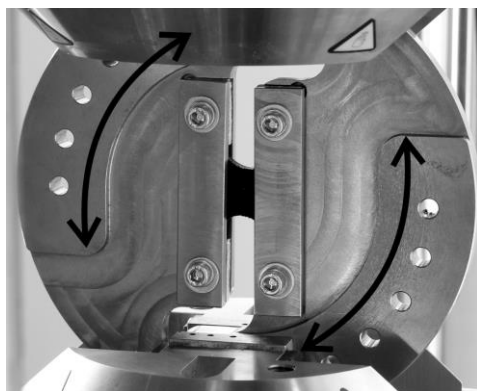


Fig. 6. Special Arcan clamps.

Three types of Arcan sample clamping were used. Firstly, thin metal plates were glued (using the Loxeal Istant 34 glue) to the Arcan sample, as shown in Fig. 7, and then the sample, together with the glued plates, was tightened using bolts and steel plates (Fig. 6). Secondly, the Arcan sample without the glued plates was tightened using bolts and steel plates. Thirdly, the Arcan sample without the glued plates was tightened using bolts and transparent plastic plates (Fig. 8). Plastic spacer sleeves with different geometry (see Fig. 7 and Fig. 8), which were printed using the Pruda i3 3D printer, were used in all three types of clamping.

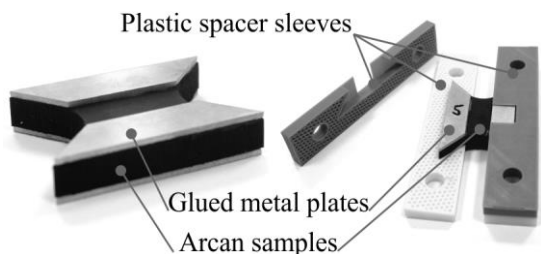


Fig. 7. Glued metal plates and plastic spacer sleeves

#### 4. Numerical simulation

The numerical model of the Arcan sample was created using the finite element method. Brick elements with 8 nodes were used. The model with highlighted surfaces where boundary conditions were applied is shown in (Fig. 9). The Arcan shear test was simulated including all three types of Arcan sample clamping. Resulting force-displacement curves were compared to the experimental curves.

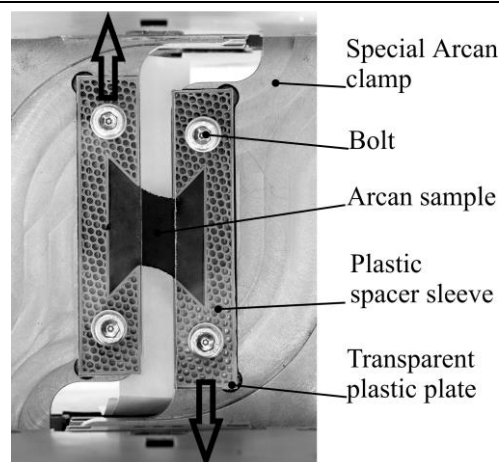


Fig. 8. Arcan sample clamped using transparent plastic plates.

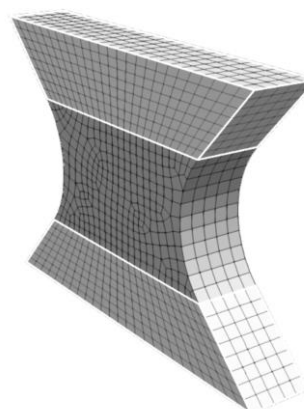


Fig. 9. Model of Arcan sample.

#### Acknowledgements

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# A NEW SURFACE PREPARATION METHOD FOR MICROSCOPIC DIGITAL IMAGE CORRELATION APPLICATIONS

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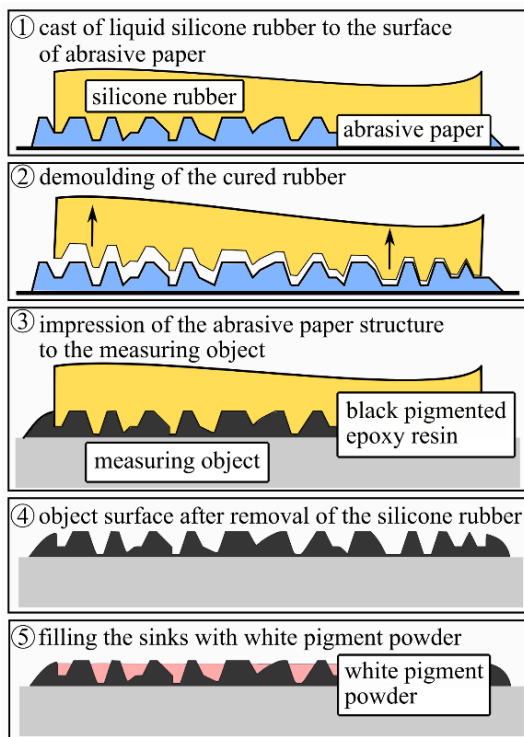
## 1. Introduction

A common field of application of experimental mechanics is the analysis of strain and displacement fields on surfaces of components. Digital Image Correlation (DIC) in particular is frequently used due to continuous improvement of image acquisition, evaluation algorithms and computing performance. The basic requirement for the usage of the method is the presence of a randomly distributed grey scale pattern, also called speckle pattern [1,2].

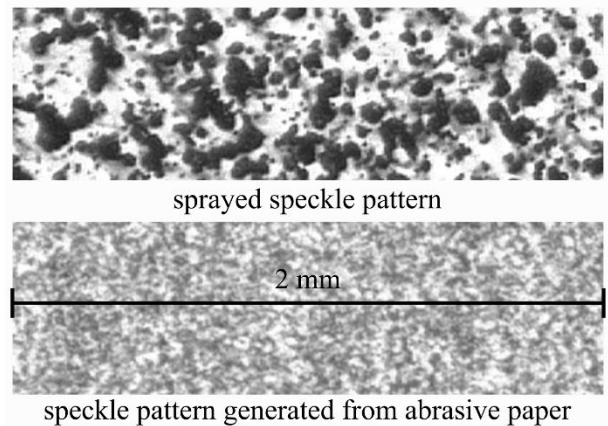
## 2. Generation of the speckle pattern

Speckle patterns are usually produced by spraying white and black paints. The smaller the field of view, the finer the speckle pattern must be to enable evaluations. Thus, it is possible that the fineness of the sprayed sample is not sufficient.

For this reason, a new preparation technique was developed, which is based on the use of abrasive paper with the grain size 800. In Fig. 1 the schematic sequence of the preparation is shown. In the first step liquid silicone rubber (Wacker Elastosil RT 601) is casted to the surface of the abrasive paper. After curing, the rubber can be easily separated from the abrasive paper and includes the profile as negative. The third step is to apply black pigmented epoxy resin to the specimen surface. With the silicone rubber negative it is possible to transmit the abrasive paper profile to the specimen. Finally, the sinks are filled with a white pigment powder. Fig. 2 shows a comparison of a sprayed speckle pattern and the above described manufacturing technique. It is demonstrated, that a fine structured pattern can be generated.



**Fig. 1.** Schematic sequence of speckle pattern preparation with abrasive paper.

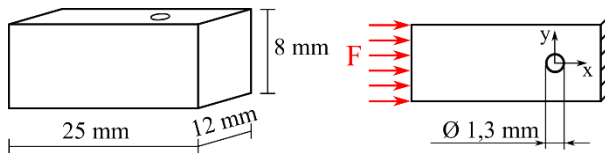


**Fig. 2.** Comparison of a sprayed and an abrasive paper generated speckle pattern.

The next section deals with the experimental test of the pattern with a small test specimen.

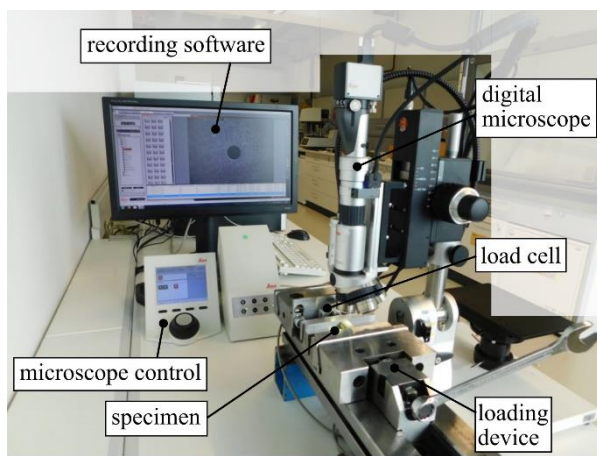
## 3. Experimental setup

For an evaluation of the new speckle pattern generation a test specimen, consisting of epoxy resin, was manufactured, see Fig. 3. A pattern was applied to the surface. Furthermore, a small hole was drilled into the specimen. Fig. 4 shows the experimental setup.



**Fig. 3.** Design of the test specimen and load case.

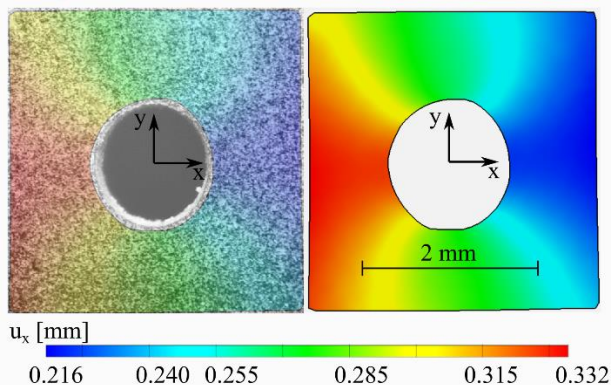
The specimen is fixed in a loading device. By means of a load cell the pressure force on the specimen can be controlled. The surface with the speckle pattern is investigated with a digital microscope (Leica VZ 700 C). The compressive force is applied manually to the specimen in several load steps. The recording software saves one picture per step.



**Fig. 4.** Experimental setup.

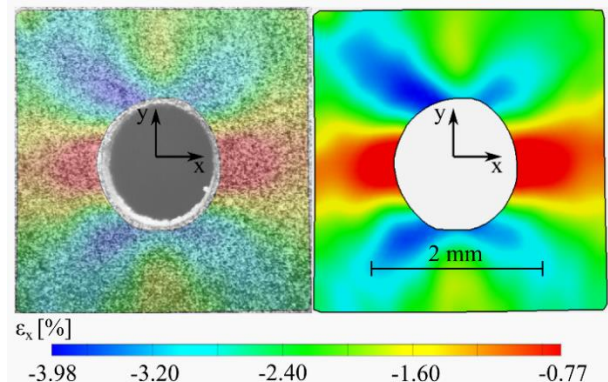
## 4. Analysis and Results

The recorded pictures were analysed with the software GOM Correlate Professional 2016. The viewed section has a dimension of 700 x 700 pixel<sup>2</sup>. For the analysis the program settings are as follows: facet size 25x25 pixel<sup>2</sup>, point distance 25 pixel, facet matching against reference step. Fig. 5 and Fig. 6 show the results of the DIC. On the left sides of the figures the speckle pattern is shown in the background.



**Fig. 5.** Displacement field around the borehole in load direction (x-axis).

The horizontal displacement field around the borehole is depicted in Fig. 5. The right edge is near to the rigid clamping; so the displacement has there its minimum. The displacement field reflects the typical behaviour. In Fig. 6 the resulting strain in horizontal direction is shown. Here a characteristic strain field around the borehole is obtained.



**Fig. 6.** Strain field around the borehole in load direction (x-axis).

## 5. Conclusions

Random speckle patterns are necessary for the DIC analysis. In a small field of view of some millimetres the pattern has to be very fine. It was shown that the new speckle preparation method based on the usage of abrasive paper is suitable for this application.

Furthermore, this procedure has the advantage, that abrasive paper is available in different grain sizes, so the speckle size can be varied. Hereinafter the speckle generation will be tested to other specimen designs and with varying grain sizes.

## Acknowledgements

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# DYNAMIC TESTING OF A SPECIMEN SETUP FOR COMBINED HIGH PRECISION UNIAXIAL TENSION-COMPRESSION TESTS OF RUBBER

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## 1. Introduction

In [1,2] a new specimen setup is introduced, which enables the investigation of phenomenological effects of rubber particular in the compression domain up to a compression strain of  $\varepsilon = -50\%$ . Note, that the specimen setup can also be used for combined or multi-hysteresis tests in the tension-compression domain. In this contribution the specimen setup is tested for dynamic investigation of rubber properties. For this, special requirements occur for the optical strain measurement.

With the new developed measurement strategy, dynamic properties of rubber like dynamic hardening, Payne-effect with storage and loss modulus can be investigated in more detail and also under high compression load.

## 2. Design of experiments

For the dynamic investigation of rubber properties, a test frequency sweep over two decades is chosen  $f = [0.1, 0.316, 1.0, 3.162, 10]$  Hz. Note, that for each frequency section 25 cycles are driven. Due to the limitation of the measuring range and prevention of imperfections [2], an optical system has to be used for the strain measurement. This restriction leads to high requirements regarding the measurement strategy. As an example, if a test frequency of 10 Hz should be applied and if a sinusoidal signal should be sampled with 20 points, a measurement frequency of 200 Hz is needed. In Fig. 1 a schematic test sequence is illustrated with adjusted measurement frequency.

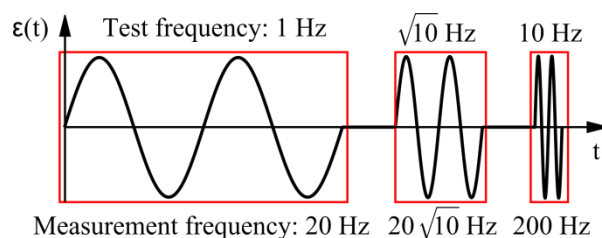


Fig. 1. Schematic test sequence.

The holding times between the sinusoidal signals is needed for switching the sample rate of the DIC measuring system.

## 3. Experimental setup

The experimental setup for the dynamic investigation of rubber properties can be seen in Fig. 2. Note, that for such dynamic tests a testing machine from INSTRON is used. The force is measured by means of a load cell and the displacement field is measured with a DIC measuring system from ARAMIS.

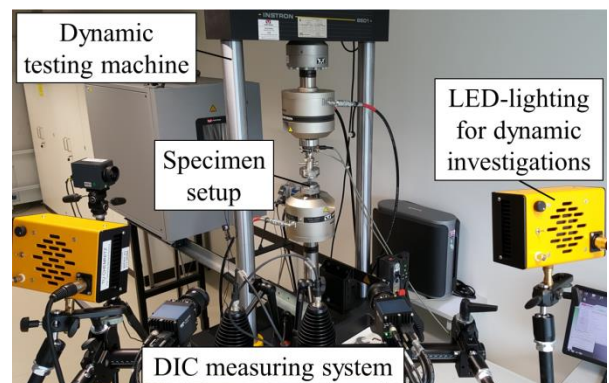


Fig. 2. Experimental setup for the dynamic investigation of rubber properties.

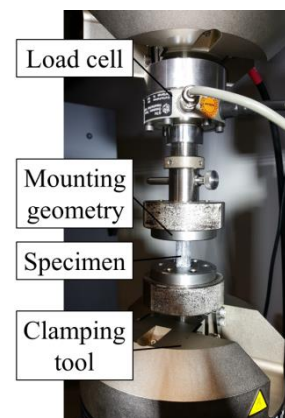


Fig. 3. Specimen setup consisting of adapters, mounting geometries and test specimen.

It is important to know, that for such high test frequencies, special requirements occur with re-

gard to the exposure time. For this, an external lighting system from Multri-LED PT is used. In Fig. 3 the specimen setup with prepared test specimen can be seen.

#### 4. Experiments and Results

The tests were performed with filled EPDM. Via Fourier analysis the storage modulus  $E'$  and the loss modulus  $E''$  can be calculated. Note, that the dynamic modulus  $E_d$  (absolute value of the complex modulus) can be achieved via Pythagoras theorem [3].

$$E_d = \sqrt{E'^2 + E''^2} \quad (1)$$

$$\tan \delta = \frac{E''}{E'} \quad (2)$$

In Fig. 4 the corresponding dynamic modulus-frequency diagram can be seen. For this first frequency sweep no preload was defined.

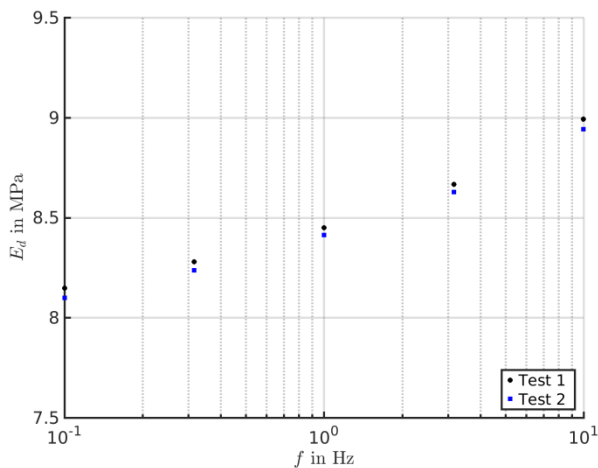


Fig. 4. Dynamic modulus with increasing frequency.

As the figure shows, the dynamic modulus is rising with increasing frequency. This effect is also called dynamic hardening. The storage modulus-frequency diagram has the same characteristic, like the dynamic modulus-frequency diagram. For the stress-strain diagram a low raise of the hysteresis can be observed, see Fig. 5.

Note, that with the new specimen setup it is also possible to investigate the dynamic hardening at a preload of e.g.  $\varepsilon = -40\%$ . It is recommended for the DIC calculation, to compare the current image with the reference image. Therefore, high strain accuracy can be achieved with the new measurement strategy. In further investigations this behavior shall be compared with different materials and preloads.

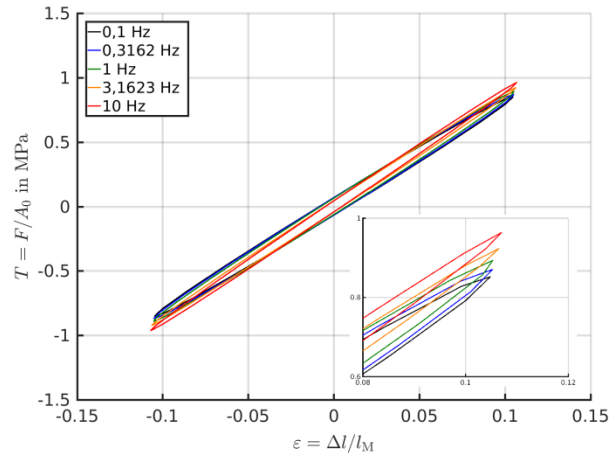


Fig. 5. Stress-strain diagram evaluated via Fourier analysis.

#### 5. Conclusions

The applied specimen setup [1,2] has a large range of applications especially in the compression domain. In this contribution, the specimen setup was successfully tested for dynamic applications. For this, the optical strain measurement was extended und tested for high measurement frequencies. As it can be seen, typical rubber properties like dynamic hardening and Payne-effect can be investigated in more detail. Finally, the developed specimen setup enables new possibilities in the field of material research.

#### Acknowledgements

The authors gratefully acknowledge the financial support from Vibracoustic GmbH & Co. KG and Freudenberg Technology Innovation SE & Co. KG.

The dynamic testing machine from INSTRON, which was used in this research, was financially supported by the German Research Foundation (DFG).

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# ELASTIC-PLASTIC CHARACTERIZATION OF 2024-T351 ALUMINUM MATERIAL BASED ON RING COMPRESSION TEST, BENDING TEST AND UNIAXIAL TENSION/COMPRESSION TESTS

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## 1. Introduction

The investigated 2024-T351 aluminum material is frequently used in finite element simulations of various cutting processes [1-3]. In most cases, the Johnson-Cook viscoplastic model is adopted to describe the material behavior at various strain-rates and at different temperature regimes. The yield stress in this model is defined as

$$\bar{\sigma} = \left[ A + B(\bar{\varepsilon}^{pl})^n \right] \left[ 1 + C \cdot \ln \left( \frac{\dot{\bar{\varepsilon}}}{\dot{\varepsilon}_0} \right) \right] \left[ 1 - \hat{T}^m \right]. \quad (1)$$

$\bar{\varepsilon}^{pl}$  denotes the equivalent accumulated plastic strain,  $A$ ,  $B$ ,  $n$ ,  $C$ ,  $\dot{\varepsilon}_0$  and  $m$  are material parameters measured at or below the transition temperature. The last term in (1) represents the temperature dependence, which is not discussed in this paper.

The accuracy of the finite element simulations strongly depends on the accuracy of the nonlinear elastic-plastic characterization of the material. It is crucial to describe the material behavior properly.

The most dominant term in definition (1) of the yield stress is the reference yield stress

$$\bar{\sigma} = A + B(\bar{\varepsilon}^{pl})^n \quad (3)$$

corresponding to the reference strain rate  $\dot{\varepsilon}_0$ . This function can be used to describe accurately the nonlinear hardening phenomenon of the investigated aluminum material. This nonlinear hardening rule contains three material parameters:  $A$ ,  $B$  and  $n$ . It is evident that parameter  $A$  represents the initial yield stress corresponding to zero accumulated plastic strain, whereas parameters  $B$  and  $n$  are used to capture the nonlinear nature of the hardening.

The characteristics of the nonlinear hardening is illustrated in the following Figure for a typical parameter set.

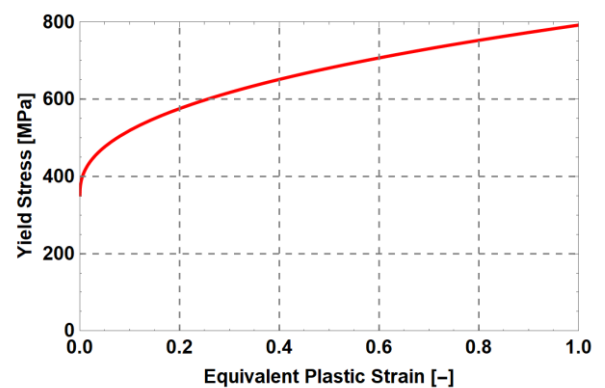


Figure 2: Stress-strain curve for parameters  $A=352$  MPa,  $B=440$  MPa,  $n=0.42$ .

## 2. Experiments

The plastic material parameters  $A$ ,  $B$  and  $n$  can be obtained by using an optimization routine for minimizing the error in between the experimental data and the model prediction. The objective function for a given experiment can be constructed by summing the squared of the differences between the predicted model response and the experimental values.

In most scientific papers, the plastic material parameters are simply obtained using the uniaxial tension test and assuming the same material behavior in the compression regime but with opposite sign. The main drawback of this method that we have no information about the real material behavior in the compression regime. Usually, the plastic behavior might be slightly different in the compression region. Consequently, the proper identification of the material parameters requires compression tests in addition to the tension tests. It is obvious, that elastic-plastic bending experiment involves tension and compression deformations simultaneously. Consequently, obtaining the

parameters from the bending test may result in more accurate description of the material behavior. The ring-compression test is similar to the bending test regarding to the fact it involves both tension and compression deformations. This type of test is also used for the determination of non-linear plastic hardening parameters [4].

A typical force-displacement curve for the ring compression test is illustrated in the following figure.

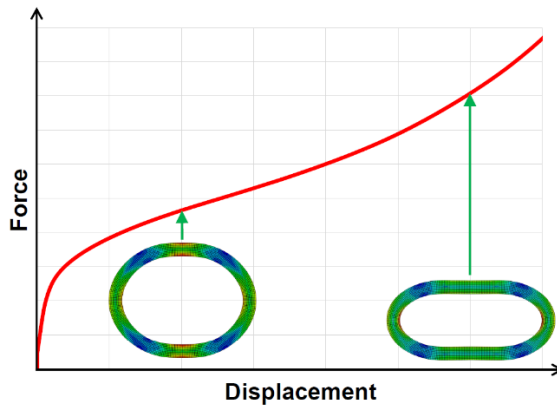


Figure 3: Force-displacement characteristics in the ring compression test

In this report, we conduct the following experiments on the investigated material:

- Uniaxial tension (UT)
- Uniaxial compression (UC)
- Bending test (BT)
- Ring compression (RC)

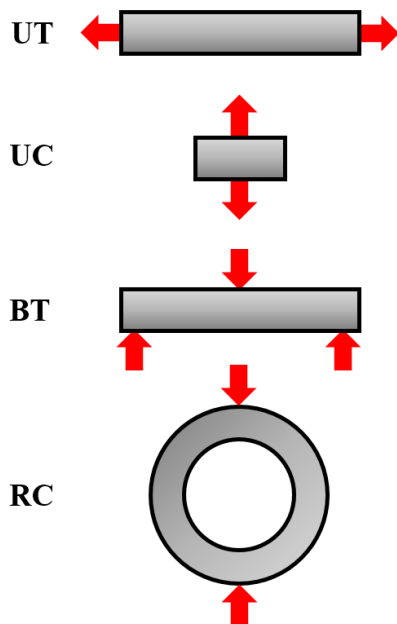


Figure 4: Different experimental tests used in the parameter fitting process

The force and displacement data are captured in each test and used in the parameter fitting process.

### 3. Parameter fitting

The overall objective function in the parameter optimization routine is constructed by

$$\Phi = \Phi_{UT} + \Phi_{UC} + \Phi_{BT} + \Phi_{RC},$$

where  $\Phi_{UT}$ ,  $\Phi_{UC}$ ,  $\Phi_{BT}$  and  $\Phi_{RC}$  are the objective functions corresponding to each experiments listed above. Consequently, the model predictions try to represent each material tests simultaneously.

Analytical solution for the bending and ring compression tests do not exist for this material model. Therefore, the model responses for the bending and the ring compression tests are obtained by finite element calculations using ABAQUS 2018 [5]. Consequently, in each iteration step in the optimization routine, a FE simulation has to be performed.

The parameter fitting algorithm is performed by linking ABAQUS with the external optimization software ISIGHT [6], in which various technique can be chosen for the minimization.

### Acknowledgements

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# INVESTIGATING THE EFFECTS OF SEVERE SHOT PEENING ON PURE IRON

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## 1. Introduction

The main advantage of using biodegradable metallic materials in biomedical engineering is eliminating the long-term problems and the need for a second surgery to remove temporary implants once they have completed their mechanical supporting task. In this regard, magnesium alloys attracted high attention; however, their diffused application has been highly limited due to their fast and uncontrolled degradation rate in biological environment [1]. Pure  $\alpha$ -grain iron is another suitable candidate in the temporary implants development; nevertheless, it shows a degradation rate that is too slow, almost comparable to permanent implants. We have considered severe shot peening that results in surface grain refinement as a potential approach to enhance mechanical performance and control the degradation rate of pure iron samples. The parameters that distinguish SSP from conventional shot peening are predominantly higher Almen intensity and surface coverage, which define the kinetic energy of the shot stream and the sample's exposure time, respectively [2, 3]. In SSP, the process parameters are set in a way to considerably increase the kinetic energy that is transmitted to the treated material.

In this study, we focused on characterizing mechanical properties of severely shot peened samples. Different sets of parameters were used with the mission to identify the most promising sets of parameters in terms of surface coverage and Almen intensity. Material characterization was performed regarding microstructure, surface roughness, microhardness and residual stresses.

## 2. Severe shot peening

Different sets of parameters were used for the surface treatment of pure iron samples, as presented in Table 1. The parameters of the applied conventional shot peening (CSP) were chosen to represent the treatment commonly used for this class of material; severe shot peening (SSP) with

much higher Almen intensity and surface coverage was considered to enhance the kinetic energy of the shot peening process in order to induce grain refinement on the top surface layer. The as-received not peened (NP) series was used as the reference. A series of samples was repeened using low Almen intensity and surface coverage just to decrease the surface roughness. Zirconia ( $ZrO_2$ ) ceramic shots were considered as peening media to avoid any possible contamination

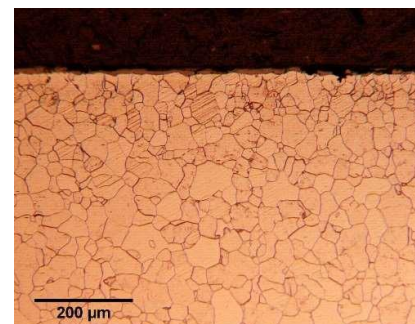
Table 1. Shot peening parameters

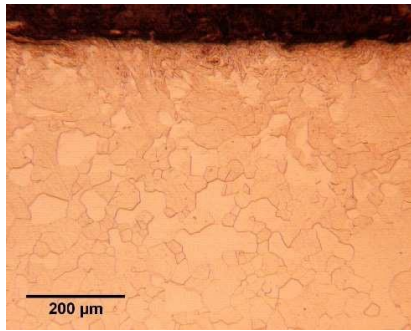
| Treatment | Almen intensity<br>[0.001 inch] | Surface<br>coverage<br>[%] |
|-----------|---------------------------------|----------------------------|
| NP        | -                               | -                          |
| CSP       | 6N                              | 100                        |
| SSP1      | 15N                             | 1000                       |
| RSSP1     | 15N                             | 1000                       |
| SSP2      | 15N                             | 5000                       |
| SSP3      | 8-10A                           | 1000                       |
| SSP4      | 14-16A                          | 2000                       |
| SSP5      | 14-16A                          | 6000                       |

## 3. Experimental characterization

Figure 1 presents the optical microscopy images of the NP and SSP5 samples; the latter shows the highest thickness for the surface affected layer with altered microstructure among all shot peened series.

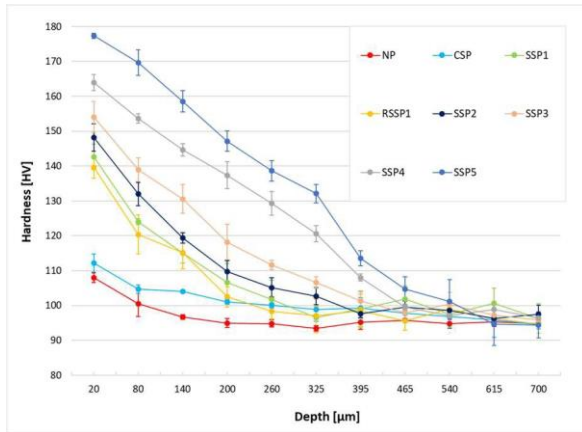
Figure 2 represents the evolution of microhardness in depth. The NP sample shows almost constant in-depth hardness along the depth. While for the shot peened series, the highest values are observed on the surface gradually decreasing to reach that of the base material.





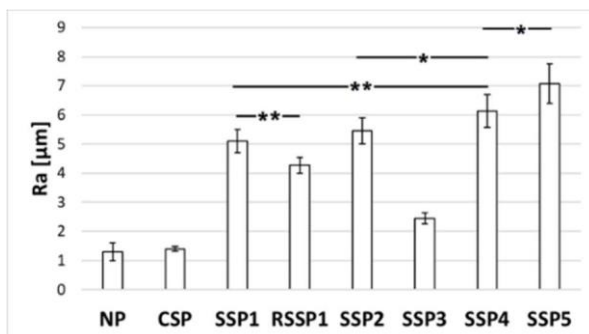
**Fig. 1.** Representative cross sectional optical micrographs of NP (top) and SSP5 (bottom) samples

The highest increase in surface microhardness is obtained for SSP5 series, which also exhibit the thickest affected layer with increased microhardness.



**Fig. 2.** Microhardness distribution on a path starting from surface along sample's cross section

Figure 3 illustrates the variation of the most widely used surface roughness parameter, i.e. arithmetic mean, for all series. SSP1 shows a sharp increase with respect to NP and CSP samples due to the sudden change in process parameters.

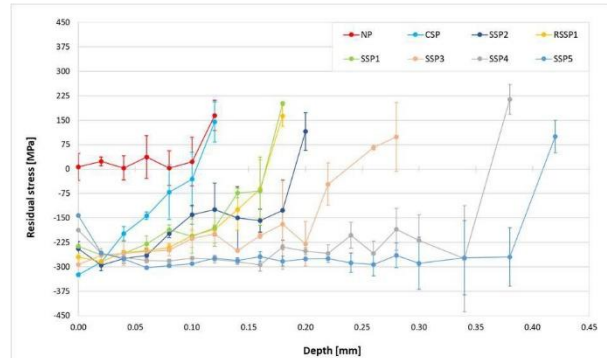


**Fig. 3.** Surface roughness parameter  $R_a$  (\*\*  $p<0.01$  and \*  $p<0.05$ )

Comparing SSP1 and RSSP1, we can see that the repeening process reduces the surface roughness. In general, increasing the kinetic energy of the process resulted in notable surface roughness

increase. The trend is not respected just by SSP3 series, caused by the much larger shot size used compared to SSP1 and SSP2.

In addition, the trend of compressive residual stresses indicate a significant increase in the depth of the layer characterized by high compressive residual stresses by increasing the kinetic energy of the process. SSP5 series are indeed showing almost 4 times higher depth affected by compressive residual stresses compared to the CSP ones



**Fig. 4.** Compressive residual stress distribution on a path starting from surface along samples' cross section

## 4. Conclusions

Severe shot peening was found to be efficient in inducing grain refinement and enhancing general mechanical properties of pure iron. The combination of the highest Almen intensity (14 - 16A) and surface coverage (6000%) was found as the most effective peening parameters to alter the microstructure, enhance microhardness, increase surface roughness and extend the depth of layer with compressive residual stresses.

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## S-N CURVE OF COMPOSITE SPECIMENS DETERMINED IN TENSION-TENSION FATIGUE TESTING

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### 1. Introduction

In this paper, the fatigue tests of composite specimen are described. Epoxy-glass composite is the material under study by using an INSTRON 8852 test machine. The specimens have been subjected by using oscillating tension-tension loading ( $R=0.33$ ) to determine the S-N curve (Wöhler curve), by applying different loads and observing how many cycles each specimen breaks for each load case.

For the fatigue tests presented in this paper, sets of five layered specimens each were made from an epoxy-fibreglass composite material. The specimens have quadratic shape, with length of 250mm and a width of 25mm. Two types of fibreglass, with different characteristics, have been used:

- a plain type fibreglass fabric ( $0^\circ/90^\circ$ ) with a thickness of 0.33mm
- a twill type fibreglass fabric ( $45^\circ/-45^\circ$ ) with a thickness of 0.2mm

The layers of the specimens were arranged so that the total thickness was approximately 2mm, which means that the specimens made from the plain fibreglass fabric had a thickness of  $6 \times 0.33$ mm, while the specimens made from the twill fibreglass fabric had  $10 \times 0.2$ mm.

The specimens have been fabricated with the dimensions according to the standard [1].

The fatigue tests have been performed with an INSTRON 8852 machine with pneumatic fatigue grips. The specimens have been subjected to time varying tension-tension loading, oscillating sinusoidal (rate  $R=0.33$ ), with the frequency of

25Hz. By using this type of charging, the autogenous heating of the specimen during the test is under  $10^\circ\text{C}$ .

According to the procedure for S-N curve drawing, a set of five specimens have been used. Each specimen was subjected to a different level of variable loading. The tests started with a high loading value for the first specimen. For each of the rest of specimens the stress level decreased..

So, the number of cycles to break has been registered.

Due to cyclic stress, the structural loading leads to micro cracks occurring in the specimens so that the loads will produce the fatigue damage ([2]). By increasing the number of stress cycles, the loading increases and in the material the damage will be accumulate, and a change in the micro - and macroscopic mechanical properties of materials will be happened. On the beginning of loading cracks appearing in the matrix fibers breakage occurs and the damage variation of material of composite specimens is not linear ([3]). Due to the breakage, the damage is rapidly developed in the material.

In this paper is described how an S-N curve has been obtained from a fatigue experiment on a set of five specimens made from a composite material was used in a few regression calculations. The calculations have been done in order to determine which of the regressions gives results that are closer to the initial curve. The regression curve that has a very small error, when compared to the initial S-N curve, can be used to determine the number of cycles it takes for a specimen to break under a certain loading, at a certain stress value.

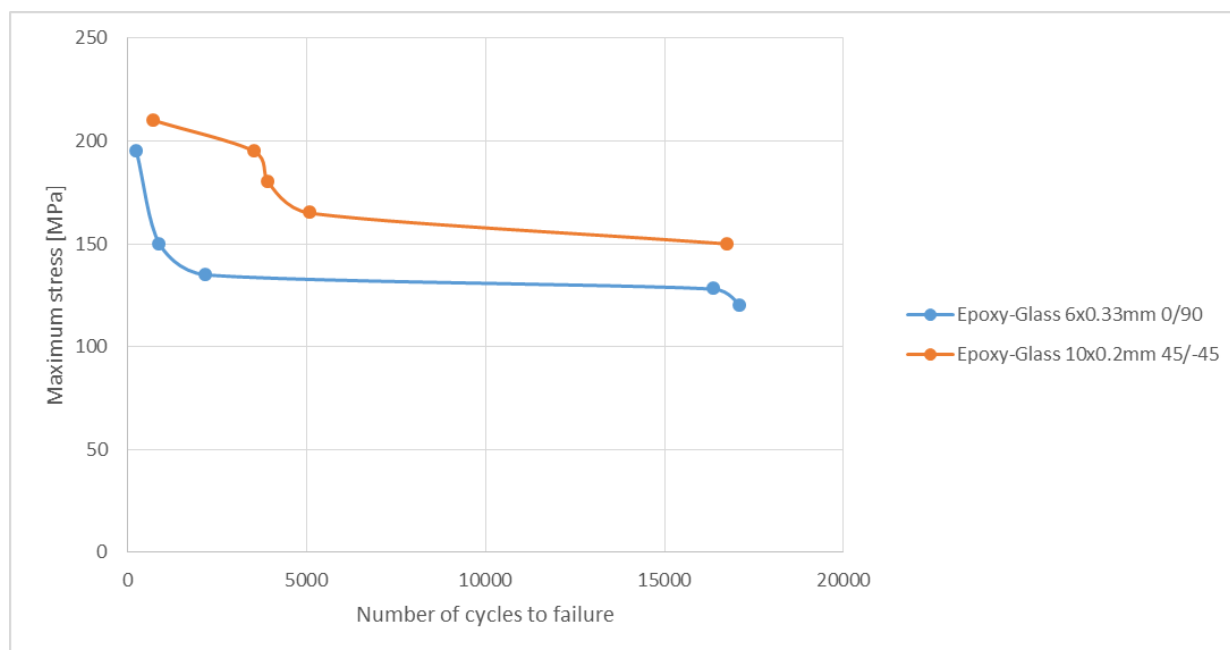


Fig. 1. S-N curves for two types of epoxy-e-glass specimens



Fig. 2. The specimen of 6x0,33mm

cycle ([3]), because fatigue machines use rotating mass that results in sinusoidal loading. In the test described in paper electro-hydraulic machine has been used that can produce a variety of waveforms, additionally to sinusoidal loading.

### Acknowledgements

The paper has been supported by Doctoral School of Mechanical and Industrial Engineering from University Dunarea de Jos of Galati, Romania.

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## 2. Conclusions

Fatigue data is usually obtained by charging the specimen with a sinusoidal tension-tension

## COMPARISONS BETWEEN SOME COMPOSITE MATERIALS REINFORCED WITH HEMP FIBERS

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### 1. Introduction

The tendency to develop biobased plastic materials has appeared because the petroleum based synthetic materials have adverse negative effects on the environment [1] like emissions during incineration, entrapment and ingestion by fish, fowl or other animals [2] and so on. Another explanation of the bioresins development is the depletion of petroleum resources. In the group of biocomposites enter biopolymers or synthetic polymers reinforced with natural fibers like: hemp, flax, jute, henequen, pineapple leaf, sisal, wood, cotton [2], nettle [3], and so on. The tensile strengths and Young modulus values of these natural fibers are generally lower than the fiber glass and are not suitable for high performance military and aerospace applications [4]. The hemp fibers are 40% cheaper than the glass fibers, have excellent moisture and mechanical stress resistance, are not attacked by moths, high tenacity (approximately 20% higher than flax) but low elongation at break, coarser compared to flax and are difficult to bleach [2]. The mechanical properties of hemp fibers are [1], [2], [5], [6]: Young modulus 26000 – 90000 MPa, tensile strength 450-900 MPa, elongation 1.6%. Other investigations were also made in [7..12].

In this paper, we have proposed a new type of biocomposite: made from a dammar based resin (60% dammar and 40% synthetic epoxy resin, inserted to decrease the hardening time) reinforced with hemp fibers with the next mechanical characteristics: tensile elasticity modulus  $E_f = 31\,000$  MPa and tensile strength  $\sigma_{fr} = 450$  MPa. For this composite, we study the static and dynamic behaviour and we make comparisons with other composites reinforced with hemp presented in engineering literature.

### 2. Experimental work

We have made rectangular composite samples reinforced with hemp fibers with the next geometrical dimensions: 170 mm length, 25 mm width and 6 mm the thickness. A general view with one of the samples is presented in fig. 1. The stress - strain curve, for a representative sample, is presented in fig. 4.



Fig. 1. A sample with dammar based resin reinforced with hemp fibers

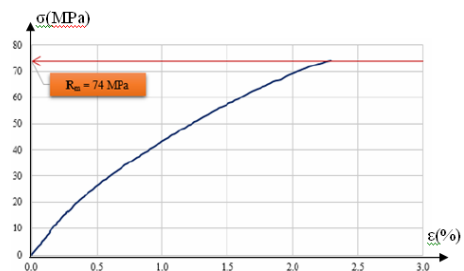


Fig. 2. The stress-strain curve for a sample reinforced with hemp fibers

For the fabricated dammar-epoxy/hemp samples, we have obtained the next mechanical properties: tensile strength between 70-77 MPa with 74 MPa as representative value; static elasticity modulus between 6133-6687 MPa with 6410 MPa as representative value; Poisson ratio between 0.49-0.55 with 0.5 as representative value; breaking elongation between 2.1-2.4% with 2.3% as representative value. In fig. 3, there is presented a representative SEM image with the sample breaking section.

The tensile elasticity modulus can be determined with (1), where  $E_f$  is the fibers tensile Young modulus,  $E_m$  is the matrix tensile modulus,  $V_f$  is the fibers volume fraction (0.34 in our study),  $n_{\parallel}$  and  $n_{\perp}$  are the fibers number on the loading direction and perpendicular to it. For our case, the fiber numbers is the same on both directions. For  $E_f = 31\,000$  MPa and  $E_m = 1812$  MPa, the theoretical value will be 6466 MPa, which is very close to the experimental result.

$$E = V_f \{ [n_{\parallel} / (n_{\parallel} + n_{\perp})] \cdot E_f - E_m \} + E_m \quad (1)$$

The composite breaking strength can be approximated with (2) when the longitudinally placed fibers take over the whole loading and the composite breakage is produced when the fibers are broken.

$$\sigma_r = [n_{\parallel} / (n_{\parallel} + n_{\perp})] \cdot \sigma_{rf} \cdot V_f \quad (2)$$

In (2) we have marked with  $\sigma_{rf}$  the fibers breaking strength. For  $\sigma_{rf} = 450$  MPa, the theoretical value will be 76.5 MPa, which is very close to the experimental result.

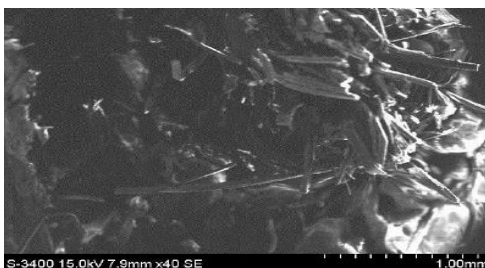
There will also be made comparisons between the new proposed biocomposites and other composites from the engineering literature made from hemp fibers.

The static behaviour of the material will be completed with a dynamic one (free vibrations, dynamic modulus, eigenfrequency, damping factor and loss factor).

### 3. Conclusions

The added value of the research made in this paper is:

- building new composite materials, having a matrix with a combination of dammar and epoxy resin (60% dammar and 40% epoxy) reinforced with hemp fibers;
- determining some mechanical characteristics from the tensile test, for the new composite, such as: tensile Young modulus, tensile strength, Poison ratio and breaking elongation;



**Fig. 3.** A representative SEM image with the breaking section for a dammar-epoxy/hemp sample

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## STATIC AND DYNAMIC MECHANICAL PROPERTIES OF A NEW GREEN RESIN BASED ON DAMMAR

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### 1. Introduction

During the last decades, researches regarding the development of biocomposites have multiplied. According to [1], the biocomposites can be divided in basic bio-materials, natural fiber reinforced composites and nano-particle reinforced composites. The first and the third refer to the materials used in biomechanics that are compatible to the human body. The second group refers to composites reinforced with natural fibers like flax, jute, sisal, kenaf, abaca, pineapple, and so on. In [1], for many natural fibers the mechanical properties are presented. The elasticity modulus and peak stress for the biodegradable composite made from poly(lactic acid) PLA with the chicken fiber feathers are also shown. A modified soy protein concentrate was presented in [2]. It has shown improved thermal stability and reduced moisture resistance compared to soy protein resin. Tensile, thermal and dynamic mechanical properties were determined for this resin and unidirectional composites with flax yarn were manufactured. Other biocomposite materials, with biodegradable resin and reinforcements, are also presented in [3,4] with some of their mechanical properties. The effects of lysine try-isocyanate (LTI) on the mode fracture energy for a hydroxyapatite particle filled poly(L-lactide)/poly( $\epsilon$ -caprolactone) biocomposite is presented in [5].

The natural resins are insoluble in water, and slightly soluble in oil alcohol or gas. From pine resins distillation there can be obtained turpentine, rosin or mastic [6].

### 2. Dynamic tests

We have made two samples with the next geometrical characteristics:

- set 1: width = 25 mm, thickness= 4 mm, mass= 49 g;
- set 2: width = 10 mm, thickness= 4 mm, mass= 22 g.

A general view with the samples sets is presented in fig. 1. The bar was clamped at one end and the other end was left free. For the two samples we used the next free lengths: 330, 310, 290, 270 and 250 mm. A schematization of the experimental montage is presented in fig. 2. We used several samples free lengths to study the damping factor per unit mass and length variation on the bar length. We recorded the vibration response with a Bruel&Kjaer accelerometer with the 0,04 pc/ms<sup>-2</sup> sensitivity. The accelerometer is connected to a NEXUS 2692-A-014 signal conditioner (Bruel&Kjaer, Denmark). The signal conditioner is connected to a data acquisition system SPIDER 8 (HBM, Germany). The data acquisition system is connected to a notebook by an USB port. In order to eliminate any possible errors inserted by the measuring system, we made a high-pass Butterworth filtration of the signal at 3Hz cut-off frequency.



Fig. 1. Samples made for tests

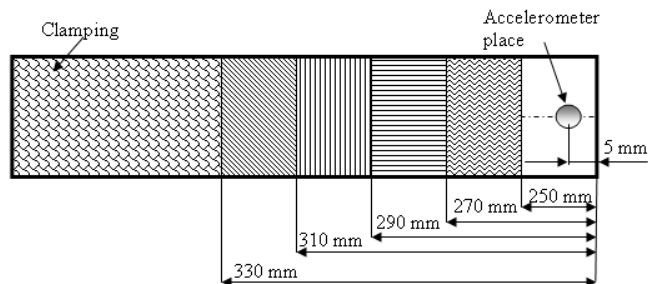


Fig. 2. Experimental montage scheme

In fig.3, the experimental recording of the free vibration for a representative sample with 25 mm width and 330 mm free length is presented. In fig. 4, the damping factor per unit mass calculus for 5 cycles is presented. The cycles are delimited by the two vertical dotted lines.

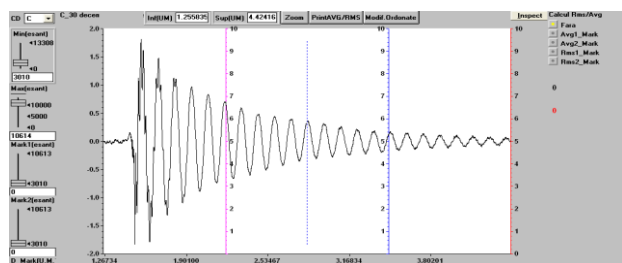


Fig. 3. Experimental recordings for a sample from set 1, with the 330 mm free length

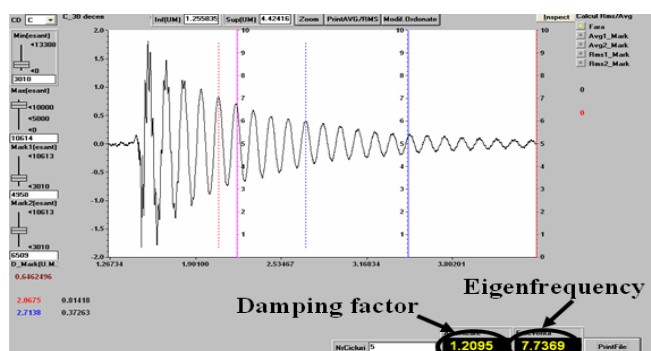


Fig. 4. The damping factor per unit mass calculus for a sample from set 1, with the 330 mm free length

All the experimental data is written in Table 1.

### 3. Conclusions

The main contribution is building a new bio resin based on dammar and studying its mechanical characteristics.

Table 1. Mechanical characteristics determined from the free vibrations recordings

| $\rho \cdot w \cdot g$<br>[kg/m] | C<br>[(Ns/m)/m] | $\eta$ | $E_{dyn}$<br>[MPa] | EI<br>[Nm <sup>2</sup> ] | set |
|----------------------------------|-----------------|--------|--------------------|--------------------------|-----|
| 0.13                             | 0.317           | 0.05   | 2222               | 0.297                    | 1   |
| 0.13                             | 0.384           | 0.053  | 2244               | 0.3                      | 1   |
| 0.13                             | 0.444           | 0.054  | 2178               | 0.291                    | 1   |
| 0.13                             | 0.55            | 0.057  | 2250               | 0.301                    | 1   |
| 0.13                             | 0.654           | 0.06   | 2145               | 0.287                    | 1   |
| 0.05                             | 0.129           | 0.049  | 2074               | 0.111                    | 2   |
| 0.05                             | 0.144           | 0.049  | 2055               | 0.11                     | 2   |
| 0.05                             | 0.154           | 0.046  | 2029               | 0.109                    | 2   |
| 0.05                             | 0.177           | 0.046  | 2044               | 0.11                     | 2   |
| 0.05                             | 0.197           | 0.043  | 2049               | 0.11                     | 2   |

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# EXPERIMENTAL STUDIES ON UNIAXIAL AND ECHIBIAXIAL TENSILE TESTS APPLIED TO PLASTIC MATERIALS SHEETS

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## 1. Introduction

The percentage of plastic materials used in different applications compared to other types of materials is increasing in the last period more and more. This is due to the many uses they have in the industry and the benefits of these plastic materials. Because also the mechanical processing methods dedicated to these materials are being varied, the present paper aims to determine the behavior of few plastic materials such as: polyamide (PA), polyethylene (PE), polyoxymethylene (POM-C) and polytetrafluoroethylene (PTFE), in cold plastic deformation.

To determine the plastic deformation behavior of these materials, we chose two types of mechanical tests, the uniaxial tensile test and the echibiaxial tensile test on the spherical punch.

## 2. Materials and methodology

To determine the mechanical characteristics using the uniaxial tensile test, we used the Instron 5587 universal test machine. To determine the behavior of the plastics sheets in the echibiaxial tensile test, we used an experimental stand (shown in Figure 1) specifically designed for this type of test. The stand is composed of the test device, the acquisition and processing system of the experimental data and the deformation measurement system. In the upper part, the test device has a modular test area fixed to the stand table and in the lower part has the hydraulic drive cylinders. The operation of the equipotential stretch test bench is a lower drive with the fixed active plate. Active elements: punch, active plate, and retaining ring; are modular, quickly changeable. In case of this stand, the punch has a vertical trajectory from bottom to top and active plate is fixed. Active elements of this device are: punch, active plate, and retaining ring; all these components are modular and quickly changeable.

To determinate the major and minor strains in specimens during the tests, was used the Aramis system which is the real-time measurement system.



**Fig. 1.** Experimental layout for the echibiaxial tests

For the second test (echibiaxial tensile test), the optical measuring system was positioned and calibrated so that the measuring area is large enough to allow the two high speed video camera of the Aramis system to fully register the test process to which the specimens are subjected.

The blank sheets used were individual blanks of square shape, with dimensions of 250 mm x 250 mm, made of, polyamide (PA 6.6), polyethylene (PE), polyoxymethylene (POMC) and polytetrafluorethylene (PTFE) and of initial thicknesses  $t = 1$  mm (PA6.6),  $t = 1.4$  mm (PE),  $t = 1.0$  mm (POMC) and  $t = 1.0$  mm (PTFE).

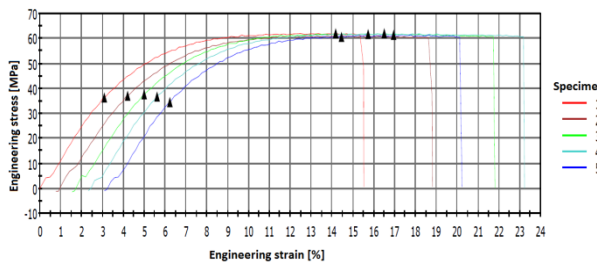
### 3. Results

The results of the uniaxial tensile tests were focused to determining the Young modulus, yield stress, engineering stress, engineering strain and tangent modulus. For each type of materials have been made and tested five. The average values for yield stress, engineering stress and engineering strain are shown in Table 1.

**Table 1.** Results for uniaxial tension tests

| Material type | Yield stress [MPa] | Engineering stress [MPa] | Engineering strain [%] |
|---------------|--------------------|--------------------------|------------------------|
| PA            | 17.5               | 41.8                     | 134                    |
| PE            | 12.6               | 24.6                     | 508                    |
| POMC          | 38.4               | 59.8                     | 18                     |
| PTFE          | 10.1               | 23.0                     | 361                    |

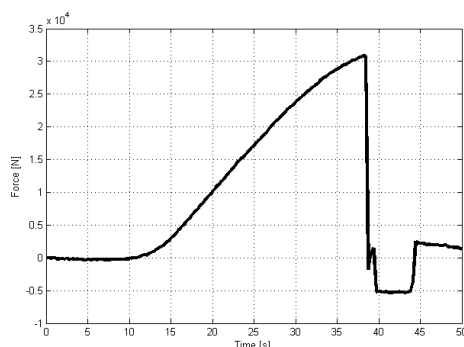
For each material, the engineering stress – strain curve for uniaxial tensile test will created. Figure 2 shows test results (engineering stress – strain curves) for all five tested specimens made from POMC material.



**Fig. 2.** Engineering stress - strain curves for uniaxial tension tests for POMC sheet

If we look on these curves we can see a very good repeatability of the results of the uniaxial tensile tests.

Regarding to the echibiaxial tensile test, the results was focused on determining the maximum height of the calotte before specimen breaks and to the maximum force at which the specimen breaks.



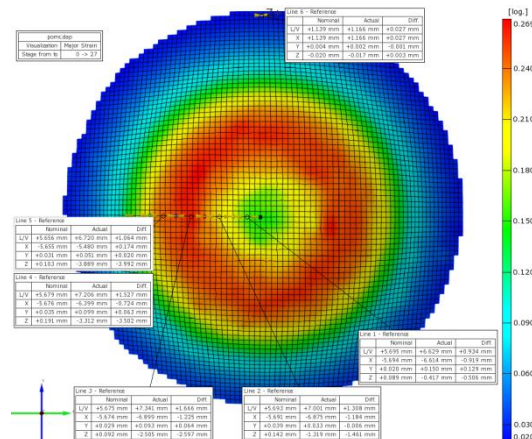
**Fig. 3.** Load vs. time curve for echibiaxial tension test for POMC sheet

Figure 3 shows the load vs. time curve for the same material type (POMC) and Table 2 presents the results of the echibiaxial tensile test for the same four types of materials like in the uniaxial tensile test.

**Table 2.** Results for echibiaxial tensile tests

| Material type | Load [kN] | Displacement [mm] |
|---------------|-----------|-------------------|
| PA            | 27.8      | 114               |
| PE            | 22.3      | 98                |
| POMC          | 30.1      | 31                |
| PTFE          | 10.3      | 57                |

At the echibiaxial tensile test, we determined using the Aramis optical system: major strain, minor strain, thickness reduction and maximum displacement before breaks. Figure 4 shows the major strain variation for POMC material.



**Fig. 3.** Major strain variation for echibiaxial tension test for POMC sheet

### 4. Conclusions

From the analysis of the results of these types of tests above, it is noted that polyamide and polyethylene have the best deformation capacity for both, uniaxial and e echibiaxial tensile test. The weakest behavior is polyoxymethylene which has a low elongation and requires large forces for deformation.

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# STATIC AND DYNAMIC TESTS FOR DETERMINING THE CHARACTERISTIC PERFORMANCE OF MECHANICAL SPLICES IN REINFORCED CONCRETE

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## 1. Introduction

Joints through Mechanical process them steel bars concrete with use mechanical coupling devices are used for realization continuity fittings different items Structural concrete reinforced, such as columns, beams, stair elements, foundations, consoles.

This process of connecting the profiled steel bars shows benefit reduction consumption steel (compared to traditional connection systems).

Mechanical Couplers can be used for items elements to fatigue, to important alternant loads or seismic type.

Couples can be of following types :

- plug with internal thread crimped a bar reinforcement with periodic profile;
- steel ribbed bar PC type, hot rolled 60, PC52 B500B, B500C, having B550B characteristics according to ST 009: 2011, forged at one end and threaded;
- end couple;
- weldable threaded couple;
- connecting rods of various types;
- transition bars;
- position couple .

## 2. Objectives

The goal this works is to present attempts static and dynamic needed for determination resistance and ductility characteristics of the mechanical couplings.

Also, 's followed check bad the terms of technical requirements according to the provisions to the standard SR ISO 15835-1 " Steel for reinforcement concrete. Locks mechanical head to head for bars. Part I: Conditions":

- resistance and ductility to static forces;
- slipping to the static forces;
- fatigue loading characteristics to a large number of cycles in the elastic field ;
- properties after a small number of reversible cycles in the elasto- plastic area.

These conditions must met to be use joints in the bracket that are subjected to fatigue, to important alternant loads or seismic type.

## 3. Experimental testing

Subject couple test It is fixed in the test equipment (Figure 1) so that loading to be transmitted axially and as much is possible, following they bending moment on all the length of the test piece to be minimal.



Fig.1.

Slip measurement must made free preloading of the test piece. If it is inevitable a small preload to secure post effort preload in the bar must be less than 4 MPa and measurement corresponding slip if it is if need noted and include in the test report.

## 4. Results

The experimental results obtained in compliance with the requirements of SR EN 13515-2: 2007 are presented in test reports, which contain all the information necessary to assess the performance of the couplings.

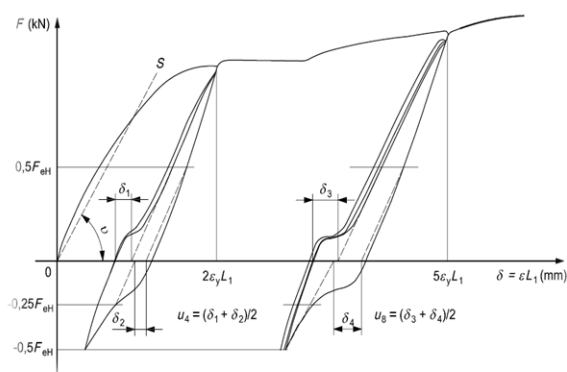


Fig.2.

## 5. Conclusions

Following their testing conducted on couplings mechanical was found as they can be used for items subjected to fatigue, to important alternating loads or seismic according to the category HLC2, in compliance with ISO 15835-1:SR 2016

Table 1 shows the mechanical characteristics of mechanical couplings with steel bars - hot rolled concrete in the traction test - high deformation compression.

Table 1

| Nr. proba | L (mm) | $\delta$ (mm) | $\delta_1$ (mm) | F (kN)    | $\varepsilon$ (mm) | $\varepsilon_1$ (mm) |
|-----------|--------|---------------|-----------------|-----------|--------------------|----------------------|
| 1.        | 220    | 1.249         | 3.13            | 33.8<br>4 | 0.210              | 0.275                |
| 2.        | 200    | 1.136         | 2.84            |           | 0.085              | 0.086                |
| 3.        | 215    | 1.220         | 3.05            |           | 0.100              | 0.160                |

Table 2 presents the mechanical characteristics of the same type of mechanical couplings with steel bars - hot rolled concrete in the traction test - high deformation compression.

Table 2

| Nr. proba | L (mm) | $\delta$ (mm) | $\delta_1$ (mm) | F (kN) | $\varepsilon$ (mm) | $\varepsilon_1$ (mm) |
|-----------|--------|---------------|-----------------|--------|--------------------|----------------------|
| 1.        | 320    | 2.060         | 5.16            | 82.76  | 0.210              | 0.275                |
| 2.        | 290    | 1.870         | 4.68            |        | 0.085              | 0.086                |
| 3.        | 295    | 1.905         | 4.76            |        | 0.100              | 0.160                |

where:

$\delta$  - traction deflection for first 4 cycles (mm);

$\delta_1$  - traction deflection for last 4 cycles (mm)

F - compressive force (KN)

$\varepsilon$  - Elongation after the first 4 cycles (mm)

$\varepsilon_1$  - Elongation after the last 4 cycles (mm) (mm)

Table 3 shows the mechanical characteristics for hot-rolled concrete,  $\Phi 25$ , having a retention limit  $Re \geq 500N / mm^2$  (according to ST 009: 2011).

Table 3

| Nr. proba | Diameter d (mm) | $F_{eh}$ (N) | Elongation to $F_{eh}$ , $\Delta l$ (mm) | L (mm) | $\varepsilon$ (-) |
|-----------|-----------------|--------------|------------------------------------------|--------|-------------------|
| 1.        | 24.56           | 280300       | 0.158583                                 | 50     | 0.00319           |
| 2.        | 24.57           | 279436       | 0.157396                                 |        |                   |
| 3.        | 24.59           | 279168       | 0.162747                                 |        |                   |

## References

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# EXPERIMENTAL DETERMINATION OF FATIGUE PROPERTIES AND E-N CURVES FOR S355J2+N STEEL GRADE

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## 1. Introduction

Today steels represent the most used group of mechanical materials. Steels are used in various branches of industry for construction of bridges, buildings, ships, cars, rail vehicles, railways. There are several thousand types of steel obtained by an appropriate combination of carbon and alloying elements of different characteristics. Because of the good mechanical properties, good cutting, forming (forging, rolling, extrusion, pressing), good weldability and low prices, structural steels are widely used in industry. The most commonly used steel for production of carrying parts of structures subjected to dynamic loads and low temperatures is medium-strength S355J2+N steel grade.

In this paper, fatigue properties and  $\varepsilon$ -N curves for S355J2+N steel grade were obtained experimentally.

## 2. Strain life ( $\varepsilon$ -N) fatigue approach

Strain life ( $\varepsilon$ -N) approach belongs to local fatigue approach and it can be used for low cycle and high-cycle fatigue. In  $\varepsilon$ -N approach loading is a combination of elastic and plastic deformation on the macro scale. In low cycle region, the plastic strain component is dominant, whereas in the high-cycle region the elastic strain component is dominant.  $\varepsilon$ -N approach uses cyclic strain-controlled tests because they better characterize fatigue behavior of a material than cyclic stress-controlled tests, particularly in the low cycle fatigue region. Mathematical model used to describe the fatigue behavior of material under cyclic strain-controlled tests to obtain cyclic stress-strain ( $\sigma$ - $\varepsilon$ ) curve is given by Ramberg-Osgood approach [1] presented by equation:

$$\varepsilon_a = \frac{\Delta \varepsilon}{2} = \varepsilon_{a,e} + \varepsilon_{a,p} = \frac{\sigma_a}{E} + \left( \frac{\sigma_a}{K'} \right)^{\frac{1}{n}}. \quad (1)$$

The total strain-life ( $\varepsilon$ - $N_f$  curve) is therefore expressed as the sum of Basquin's and Manson-Coffin's part by equation [2]

$$\varepsilon_a = \frac{\sigma'_f}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c. \quad (1)$$

## 3. Strain controlled uniaxial tension-compression fatigue test

Uniaxial tension-compression fatigue tests of the S355J2+N steel grade were carried out according to the internal procedures of the Centre for engineering software and dynamic testing at Faculty of Engineering University of Kragujevac, based on the ASTM E468-90 [3] and ASTM E606-92 [4] standards.

Uniaxial tension-compression fatigue tests were performed by applying a sinusoidal wave in an universal servo-hydraulic machine SHIMADZU Servopulser EV101K3-070-0A. All specimens have been finely polished to minimize the surface roughness effects. The tests were strain controlled by means of a SHIMADZU DYNASTRAIN TCK-1-LH dynamic extensometer with a  $\pm 1$  mm working range.

The strain levels used to control the fatigue tests were chosen from the previously performed monotonic material characterization. The selection criteria was to obtain fatigue life cycles between  $10^4$  and  $10^6$  cycles. The uniaxial tension-compression test planning (loading ratio  $R=-1$ , i.e. mean strain amplitude  $\varepsilon_m=0\%$ ) is: five levels, three repetitions per level with a range of strain amplitude from 0.15% to 0.20%. The test frequency used in the characterization was in the range of 3–10 Hz, and the crack initiation criterion (failure criterion) was the quick stiffness loss (load amplitude loss of about 10%).

## 4. Results and discussion

Table 1 shows the results of the uniaxial tension–compression strain-controlled fatigue tests. The normal stress amplitude  $\sigma_a$  was calculated by means of the maximum load applied to the medium life cycle in stable conditions divided by cross sectional area of the specimen at the beginning of the test.

**Table 1.** Experimental uniaxial tension–compression strain-controlled fatigue test results of S355J2+N steel grade

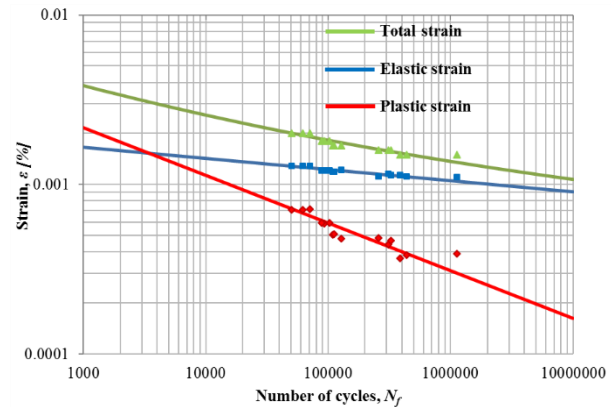
| Specimen number. | $\varepsilon_a$ [%] | $\sigma_a$ [MPa] | $N_f$   |
|------------------|---------------------|------------------|---------|
| 1-1              | 0.20                | 271.2            | 61900   |
| 1-2              | 0.20                | 270.5            | 50300   |
| 1-3              | 0.20                | 269.4            | 71000   |
| 2-1              | 0.18                | 254.1            | 92600   |
| 2-2              | 0.18                | 253.2            | 88300   |
| 2-3              | 0.18                | 253.2            | 103300  |
| 3-1              | 0.17                | 256.5            | 127600  |
| 3-2              | 0.17                | 250.4            | 109600  |
| 3-3              | 0.17                | 250.1            | 116000  |
| 4-1              | 0.16                | 243.1            | 310700  |
| 4-2              | 0.16                | 237.7            | 324700  |
| 4-3              | 0.16                | 234.3            | 257700  |
| 5-1              | 0.15                | 234.2            | 437500  |
| 5-2              | 0.15                | 233.0            | 1117200 |
| 5-3              | 0.15                | 238.2            | 389200  |

Based on the results shown in Table 1. and statistical analysis according to standard ASTM E739-91(2004) [5], uniaxial tension–compression strain controlled mechanical properties of S355J2+N steel grade have been shown in Table 2.

**Table 2.** Uniaxial tension–compression strain controlled mechanical properties of S355J2+N steel grade

| Uniaxial cyclic properties                      | Value       |
|-------------------------------------------------|-------------|
| Cyclic strength coefficient, $K'$               | 1470.45 MPa |
| Cyclic strain hardening exponent, $n'$          | 0.2344      |
| Cyclic yield strength, $\sigma_y'$              | 342.65 MPa  |
| Fatigue strength coefficient, $\sigma_f'$       | 575.25 MPa  |
| Fatigue strength exponent, $b$                  | -0.0656     |
| Fatigue ductility coefficient, $\varepsilon_f'$ | 0.0182      |
| Fatigue ductility exponent, $c$                 | -0.2087     |

Based on experimentally obtained uniaxial tension–compression strain controlled mechanical properties of S355J2+N steel grade, strain–life curve (log–log representation) has been determined and shown in Figure 1.



**Fig. 1.** Strain–life curve from uniaxial tension–compression strain-controlled fatigue test of S355J2+N steel grade

## 5. Conclusions

This study has presented an experimental determination of fatigue properties and  $\varepsilon$ – $N$  curves for S355J2+N steel grade. The hysteresis loop behavior, cyclic stress–strain response and strain–life curve were determined by testing smooth specimens. A complete cyclic characterization of the material is obtained, including the new experimental  $\varepsilon$ – $N$  fatigue curves for uniaxial tension–compression strain-controlled fatigue test. In order to determine fatigue properties of material S355J2+N, the testing procedure was performed with the help of special measurement device, servo-hydraulic SHIMADZU Servopulser EV101K3-070-0A.

## Acknowledgements

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## EXPERIMENTAL DETERMINATION OF STRAIN HARDENING EXPONENT FOR STRENGTH 700

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### 1. Introduction

The most common material today in all aspects of daily life is steel. Its history is very rich, but it can be said and the future, although we are witnessing of increasing usage of new materials in the present. Many world grooving economies, as well as those of the future, owe their progress to a large part of the development and consumption of steel and its products. The development of steel has created an opportunity for manufacturers to produce a wide range of steels with different characteristics to suit the intended use, by combining the small percentage quantity of carbon with other alloying elements. The choice of adequate material for projected part depends on its ability to respond to the established requirements regarding the functional durability to the load it is exposed to, as well as the possibility of making them according to the given dimensions.

During certain technological processes of production of parts and exploitation in extreme conditions like high temperatures, knowing the behavior of the material and its mechanical characteristics in these conditions is necessary. For ductile materials, reduction in strength and additional increases ductility are coming with increasing temperature. This behavior of the materials is the result of the influence of the temperature on its deformation. In order to determine the characteristics of the material in this case, it is necessary to perform material testing at elevated temperatures

In this paper, material characteristic of high strength steel STRENGTH 700 was studied experimentally at elevated temperature.

### 2. Testing at elevated temperature

The tests at elevated temperature consist in assessing the behavior and determining the

mechanical properties of the material under load. Three basic test methods are distinguished [1]:

- Short-term heating (tensile testing, compressing, bending, impact strength, etc.),
- Long-term heating (creeping, relaxation, etc.),
- Short-term and long-term heating after long-term heating at elevated temperatures.

Tensile testing at elevated temperatures over 35 °C is carried out in accordance with a procedure that basically coincides with the test procedure standard for tensile at room temperature. The methodology for testing of metallic materials at elevated temperatures is defined by the standards ISO 6892-2 [2], ASTM E21-17 [3]. The simplest form of the equation of the hardening curve in the case of uniaxial tension, without influence of temperature and strain rate, can be written in the form [4]

$$\sigma = K(\varepsilon_p)^n \quad (1)$$

where  $n$  is the strain hardening exponent and  $K$  is coefficient of proportionality. A much more adequate form of equation for strain hardening is variation of Ramberg-Osgood equation in form [5]

$$\sigma = \sigma_{uy} + C_y(\varepsilon_p)^{n_y} \quad (2)$$

in which  $C_y$  and  $n_y$  represent material constant.  $\sigma_{uy}$  is yield stress. These material constants can be determined from the expressions

$$C_y = \frac{R_m e^n - R_p}{\frac{R_m e^n}{n} - R_p} \quad (3)$$

$$n_y = \frac{R_m e^n}{R_m e^n - R_p} n.$$

in which  $R_p$  is yield stress and  $R_m$  is ultimate strength.

### 3. Methodology of tensile testing at elevated temperatures

Uniaxial tension at elevated temperature was carried out at the Center for Engineering Software and Dynamic Testing at the Faculty of Engineering Sciences, University of Kragujevac, according to standard ISO 6892-2 [2]. The testing program includes the testing of mechanical characteristics at five temperatures from 100 °C to 500 °C, including room temperature. By optical scanning of the samples after fracture and by placing separate parts in contact by matching the fractures surfaces, a digital reconstruction of the specimen geometry was performed during the test at the time of the breakage. On that way, the actual fracture deformations were determined.

All tests were performed on universal servo-hydraulic machine SHIMADZU servopulser EV101K3-070-0A. For measuring of strain at high temperature, extensometer EPSILON 3548-025M-050-ST was used which was installed in the furnace in a horizontal position. The selected strain rate is  $5 \cdot 10^{-4} \text{ s}^{-1}$ , which corresponds to the velocity of the cross head of 0.02 mm/s. Optical measurements were realized using the optical measuring device ATOS COMPACT SCAN 5M, which basically represents a contactless measuring system for digitizing the complete geometry of visible surfaces on objects of different sizes and complexities. Measuring volume of optical device was 300/MV300.

### 4. Results and discussion

The summarized results are shown in the diagram in Figure 1.

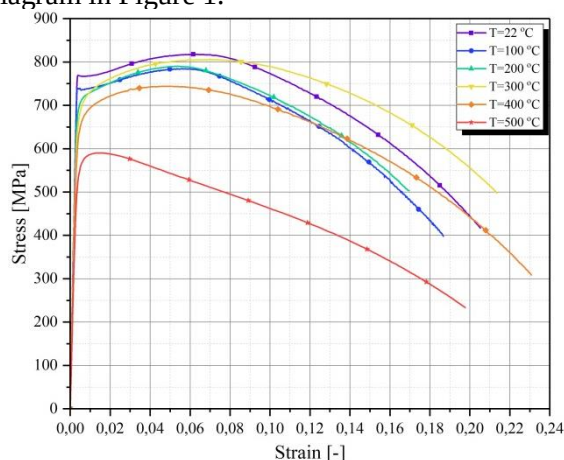


Fig. 1. Comparative overview of test results for each temperature

It is concluded that at temperatures exceeding 400 °C there is a sudden drop in the strength of

materials for both types of test tube. Also, it is noticeable that at temperatures of 200 °C and 300 °C the values of the stress are greater than the value corresponding to the room temperature. The temperature range in which this phenomenon occurs is known as the *blue brittle* region. Table 1 shows experimental determined hardening parameters.

Table 1. Experimental determined hardening parameters for both form of equation for STREN X700

| T<br>[°C] | $\sigma = K \varepsilon^n$ |              | $\sigma = \sigma_{yy} + C_y \varepsilon^{n_y}$ |                |        |
|-----------|----------------------------|--------------|------------------------------------------------|----------------|--------|
|           | $n$                        | $K$<br>[MPa] | $\sigma_{yy} = R_e$<br>[MPa]                   | $C_y$<br>[MPa] | $n_y$  |
| 22        | 0,05094                    | 992,67       | 767,38                                         | 378,1          | 0,4708 |
| 100       | 0,04616                    | 936,61       | 735,68                                         | 333,7          | 0,4401 |
| 200       | 0,05600                    | 975,84       | 710,53                                         | 367,6          | 0,3742 |
| 300       | 0,06277                    | 1015,74      | 692,84                                         | 407,3          | 0,3259 |
| 400       | 0,05008                    | 908,64       | 657,15                                         | 319,4          | 0,3132 |
| 500       | 0,01893                    | 652,04       | 559,30                                         | 123,0          | 0,2699 |

### 5. Conclusions

This paper has presented an experimental determination of strain hardening parameters for high strength steel STRENX 700 at elevated temperatures. By increasing the temperature, the material characteristics like strength and elongation are changed, especially at 400 °C and higher temperatures. The testing procedure was performed using of special measurement device, servo-hydraulic SHIMADZU servopulser.

### Acknowledgements

The authors gratefully acknowledge partial support by Ministry of Education, Science and Technological Development, Republic of Serbia, Grant TR32036.

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# DETERMINING THE EMISSIVITY FACTOR OF THERMOPLASTIC MATERIALS FOR A THERMOPLASTIC BENDING PROCESS

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## 1. Introduction

As part of the project “Development of a 3D-bending machine” the topic thermography of surfaces is essential. The contact free measurement method performs measurements by absorbing thermal radiation from the object. Other advantages include better in-process performance based on computer compatibility and the flexibility of thermographic devices. The contact-less measurement property is appropriate for measuring the surface temperature of the thermoplastic material during processing it.

To accurately detect an object’s surface temperature, the material and surface-specific emissivity factor  $\varepsilon$  must be determined. According to Eq. (1),  $\varepsilon$  is defined as a factor that compares the specific radiation of a real object (“gray body”) to a black body at the same temperature. In a case where the evaluated object is a black body,  $\varepsilon$  is equal to 1. Otherwise,  $\varepsilon < 1$  (gray body).

$$\varepsilon = \frac{M(T)}{\sigma * T^4} \quad (1)$$

Referring to Eq. (1),  $M(T)$  is the thermal emissivity of the observed object depending on its temperature ( $W/m^2$ ),  $\sigma$  is the Stefan–Boltzmann constant, and  $T$  is the temperature of the black body (K), which is equal to the observed temperature. Thermographic devices (e.g., infrared cameras) detect  $M(T)$  and calculate the surface temperature with  $\varepsilon$  as the input [1].

## 2. Determine Emissivity Factor

To determine the emissivity factors, thermoplastic samples (compare Fig. 1 data sheets in [2]) are measured using an infrared camera (Optris PI400, Optris GmbH, Berlin, GER; properties in [3]) and contact thermometer (Testo 925, Testo GmbH, Vienna, AUT) for reference

purposes. That is to say that the determined emissivity factors only apply for those specific types of samples. Any changes in surface condition (e.g., raw to smooth) require new determinations even if the material is the same.



Fig. 1. Observed samples

First, the samples are prepared with two drilled holes, as shown in Fig. 2. The larger one ( $d_1 = 3 \text{ mm}$ ,  $l_1 \gg d_1$ ) is needed to realize a black body. According to [1], this can be achieved by drilling a hole with a depth that is several times its diameter. The second hole is for measuring the core temperature with an ordinary contact thermometer for reference. A small difference between the diameters of the holes and thermometer is essential for precise measurement.

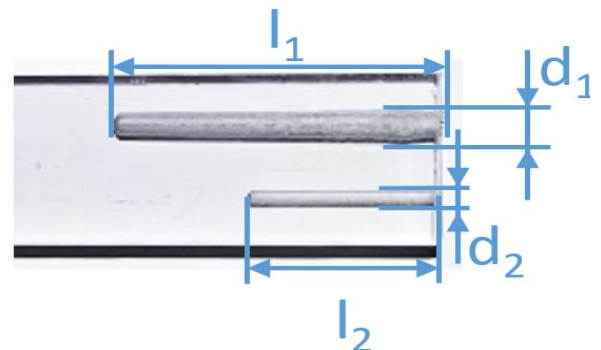


Fig. 2. Preparation of samples.

Fig. 3 shows the surface of one sample, as seen by the infrared camera. The two bright spots in this

figure correspond to the holes in Fig. 2. The right cavity shows that the black body has a higher thermal radiation than its surroundings. This is due to the ability of a black body to absorb incident thermal radiation completely, with  $\varepsilon = 1$ , because of Kirchhoff's law [1].

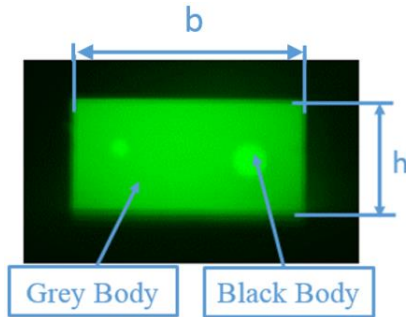


Fig. 3. Surface of the sample through infrared camera:  $b = 20$  mm,  $h = 10$  mm.

Using this knowledge, one measuring point of the processing program evaluates the temperature of the black body with a dedicated  $\varepsilon$  of 1 (compare Fig. 4). The second measuring point evaluates the mean surface temperature, excluding the two holes. To achieve the same temperature as the first measuring point (thermal equilibrium) and to determine the specific emissivity factor,  $\varepsilon$  of the second point is modified until the two temperatures are equal [4].



Fig. 4. Measuring surface temperature.

### 3. Results

Fig. 5 provides an excellent overview of the measured temperatures.

The first two columns represent the surface and black body temperatures evaluated by the infrared camera. The next two columns represent the core and surface reference temperatures measured by the contact thermometer.

As can be seen in Fig. 5, the temperatures are almost equal to one another. Any deviations are due to inaccuracies in the measuring devices and are small enough to be ignored. Table 1 shows the determined emissivity factors. Column RE describes the relative error, evaluated from the mean

surface temperature and absolute temperature of the black body (see Fig. 4).

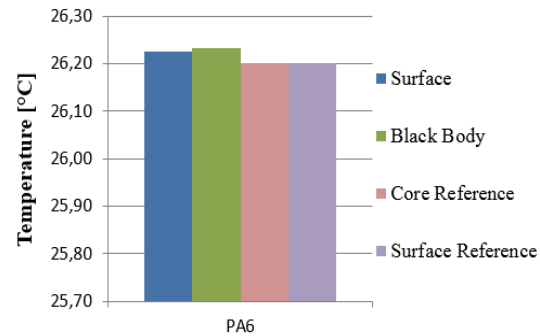


Fig. 5. Measurements of PA6.

Table 1. Emissivity factor  $\varepsilon$  and relative error of the samples

| Sample | $\varepsilon$ | RE [%] |
|--------|---------------|--------|
| ABS    | 0,965         | 0,73   |
| PA6    | 0.965         | 0,02   |
| PC     | 0,975         | 0,15   |
| PE3    | 0,96          | 0,46   |
| PE10   | 0,975         | 0,56   |
| PP     | 0,965         | 0,28   |
| PVC    | 0,965         | 0,25   |

### 4. Conclusion

As seen in Table 1, the relative error is rather low, which allows continued use of the determined emissivity factor for evaluating the surface temperature of the sample. In addition to the accurate observation of the heating process using the IFT's prototype,  $\varepsilon$  may support the selection of materials intended for thermoforming. This is owing to Kirchhoff's law, which says that the absorption of thermal radiation is equal to  $\varepsilon$ .

### Acknowledgements

The authors appreciate the financial support of the Austrian Research Promotion Agency (FFG) for the project "Clever cast and i-Cont" (849517).

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## THREE-POINT BENDING BEHAVIOUR OF ORGANIC MODIFIED COMPOSITES

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### 1. Abstract

The use of the combination of an organic agent (gelatin and wheat gluten) represents a challenge in order to obtain modified composites with superior properties compared to the unmodified epoxy system (unmodified polymer) [1]. Currently, the phenomenon of modification of the epoxy system with various types of organic and/or inorganic modifying agents is analyzed. The purpose of this research was to evaluate and analyze the three-point bending behaviour of organic modified composites [2]. Three different concentrations of a modified agent (gelatin and wheat gluten) were used in the formation of the composite materials. In the process of determining the three-point bending behaviour of the formed materials, there was also observed the mode of dispersion of the modifying agent and the interface between the modifying agent and the polymer [3]. Characterization of three-point bending behaviour was performed by comparing unmodified epoxy system and modified materials [4], [5].

Composite materials modified with various types of organic/inorganic agents as well as composite materials reinforced with carbon fiber, glass, kevlar, natural fibers, etc. are increasingly used in different production processes to obtain products that meet consumer requirements. It is also desirable to completely replace classical materials with composite materials because these modern materials have a number of advantages: specific weight, operating performance, but also a number of disadvantages, especially in the case of composite materials reinforced with fabrics: limitation of the use of reinforced materials in

certain fields of activity, and adhesion between fibers and matrices [6], [7], [8].

### 2. Experimental section

The following types of epoxy resins: Epoxy RESIN C, Epyphen RE 4020, Epoxy Resin HT, have been used to carry out this study because they have a very low shrinkage coefficient, unlike other types of resins. For each type of resin there is a certain type of hardener: Hardener C, Epoxy Hardener DE 4020 and Hardener HT. The modification of the epoxy system consisted in the introduction of a quantity of the agent (gluten and gelatin) in the proportion of 1%, 3% and 5% each. The resulting mixture was homogenized with the magnetic stirrer for 24 hours at 60° C and a mixing speed of 500 rpm. In order to determine the three-point bending behaviour of formed composite materials were performed cylindrical specimens according to SR EN ISO 14125.

The tension state of the specimens is inhomogeneous, both due to the variation of the bending moment from one cross-section to the other, and due to the fact that for a given cross-section the stresses vary linearly in the elastic range. The purpose of these tests was to observe the influence of the modifying agent on the three-point bending behaviour.

In order to characterize the three-point bending behaviour of organic modified composites a comparative study was carried out between the unmodified epoxy system and the modified materials (Fig. 1) where a critical parameter in the elastic domain was chosen, namely the elastic modulus also known as the Young's modulus.

### 3. Results and discussion

Of all the tested materials, atypical behaviour is represented by HT2 material. It contains an intermediate percentage of the modifying agent from the three concentrations used (2%). The explanation of this behaviour lies in the impossibility of homogeneous dispersion of the organic agent in the volume of the polymer and implicitly the lack of a uniform and continuous interface between the modifying agent and the polymer. If a comparison is made between the other types of formed materials, we can say that the choice of the modifying agents (wheat gluten and gelatin) was a successful one. This aspect offers the possibility to continue the research in testing these materials.

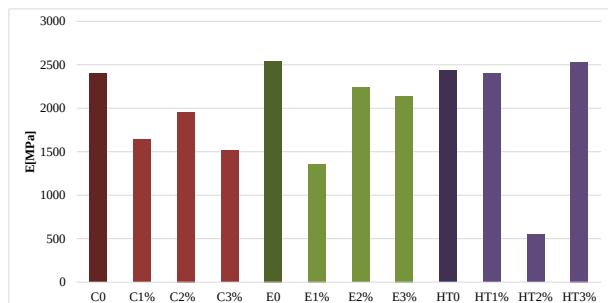


Fig. 1. Graphic representation of the elastic modulus.

### Acknowledgements

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## **SECTION III: BIOMECHANICS**



# EVALUATION OF HUMAN KNEE FLEXION-EXTENSION ON INCLINED TREADMILL USING WEARABLE SENSORS

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## 1. Introduction

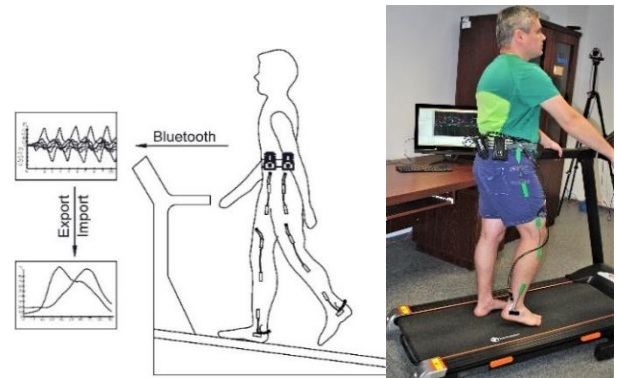
Nowadays, the importance of measuring and analyzing gait parameters has increased and it is more and more recognized and used in biomechanics, in robotics, in rehabilitation and in clinical research field, if they are accurately measured [1-4]. In the medical field, the knowledge of gait characteristics, the monitoring and evaluating changes in human gait reveal important information about quantitative objective measurement of the different spatial and temporal gait parameters, about the evolution and early diagnosis of different diseases and about physical therapy, involving rehabilitation, which helps to improve the walking function [1-4].

The objective of this study is to measure the variation of flexion-extension angles of the human knee joints while the subjects performs walking on a treadmill (TM) with different incline angles.

## 2. Experimental study

The experimental method which allows obtaining the kinematic parameters diagrams for the human knee joint uses a Biometrics Ltd data acquisition system based on electrogoniometers, which are wearable sensors successfully used in biomechanics and medical applications [4-6].

For data acquisition six electrogoniometers, one for each of six joints of both lower limbs, connected with two 8-channel DataLOG at a frequency of 500 Hz (MWX 8 Biometrics Ltd) were used. The data were transferred from DataLOG to computer in real time via Bluetooth communication. In Fig. 1 the data acquisition system mounted on the subject is shown.



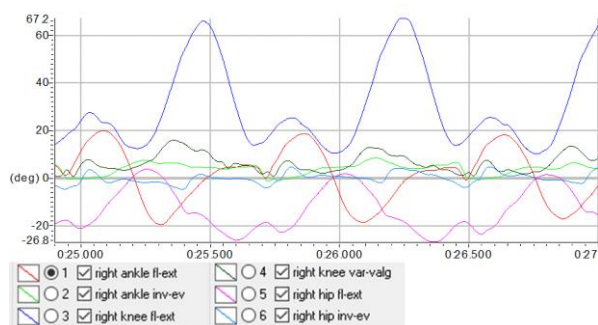
**Fig. 1.** The acquisition schema and the subject with acquisition system mounted on him

Measurements were performed on a sample of 5 healthy subjects. They were pain-free and had no evidence or known history of motor and skeletal disorders or record of surgery to the lower limbs. Each of five subjects performed un number of 5 tests on Treadmill, with a speed equal to 7.5 km/h, and different incline angles: Test 1 - 0°; Test 2 - 3°; Test 3 - 7°; Test 4 - 11°; Test 5 - 15°.

## 3. Results

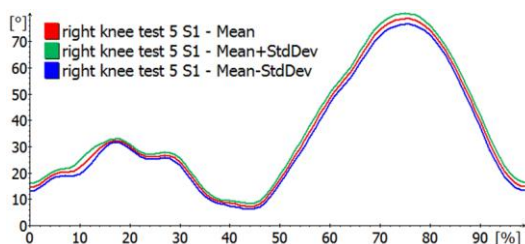
The angular amplitudes of human knee flexion-extension during the gait on the treadmill were obtained for each test as data files. In Figure 3 the diagram of experimental flexion-extension joint angles, in sagittal plane, for the three joints of left leg of Subject 1 in respect with time [s] obtained by Biometrics system, are presented.

Human gait variability from one cycle to another for the same subject and from one subject to another imposes the normalization of gait cycles. Normalization is done using SimiMotion software where the data files are transferred [5, 6].

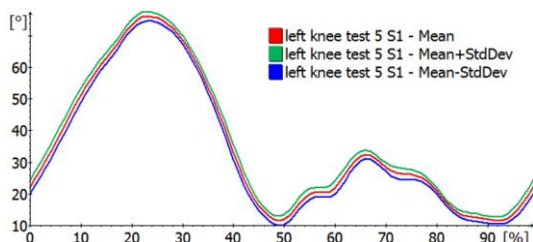


**Fig.2.** Experimental flexion-extension angles and rotation angles in frontal plane, for the right leg of subject 1.

To compare results, it is necessary to determine the average cycle for each acquired data file containing more than 30 consecutive cycles. Then, an average cycle of the entire sample of five subjects will be obtained for each of both knee joints for every test, as, for example, are shown in Fig.3 and Fig.4 for Subject 1, test 5.

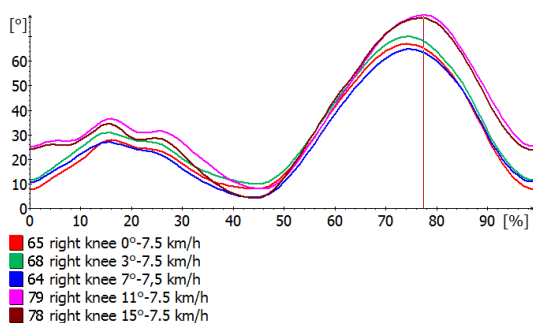


**Fig.3.** Diagrams of average cycle of right knee joint for subject 1, for Test 5.



**Fig.4.** Diagrams of average cycle of left knee joint for subject 1, for Test 5.

In fig. 5 the diagrams of average cycles of entire sample for all tests for the right knee are shown.



**Fig.5.** The diagrams of average cycles of entire sample for all test (right knee).

Based on these diagrams, an analysis of the influences of the incline angle of the treadmill on the amplitudes of flexion-extension knee angle during walking on treadmill for a constant speed, could be done.

In table 1 are present the maximum value of average cycle.

**Table 1.** The Maximum value of average cycle.

| Test name | Subj.1 | Subj.2 | Subj.3 | Subj.4 | Subj.5 | Sample |
|-----------|--------|--------|--------|--------|--------|--------|
| Test 1    | 60     | 69     | 67     | 64     | 65     | 65     |
| Test 2    | 66     | 68     | 69     | 66     | 68     | 67     |
| Test 3    | 61     | 69     | 67     | 62     | 64     | 65     |
| Test 4    | 75     | 82     | 82     | 76     | 80     | 79     |
| Test 5    | 74     | 81     | 80     | 77     | 79     | 78     |

## 4. Conclusions

A study based on the tools of gait analysis and on a Biometrics data acquisition system based on wearable sensors, investigates the human flexion-extension movements of the human knee joints during walking on plane and inclined treadmill with different incline angles. The main gait parameters and the average cycle and parameters of entire sample are obtained for both knees (right and left) of each subject for all performed tests. A comparison between the values of main gait parameters is done. For future work, numerical simulation and experimental results could be used to design and develop human-inspired robotic structures used in medical, assistive or rehabilitation fields.

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# EXPLORING EXPERIMENTALLY THE POTENTIALITIES OF PLATELET RICH PLASMA IN TREATING PELVIC ORGAN PROLAPSE

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## 1. Introduction

It is known from human anatomy that the supportive connective tissue of female uterus is divided into 3 levels, i.e., that of uterosacral (USL) and cardinal ligaments, that of the levator ani and that of the perineal body. Changes in these structures result to pelvic organs prolapse (POP) into the vagina. It is mentioned that the lifetime risk of surgery for POP in the general female population approaches 11%. Over 30% of women over the age of 50 years suffer from POP [1]. In the United States almost 200,000 women undergo surgery for POP annually, while the surgical correction of POP is anticipated to increase by about 45% during the next 30 years.

As possible causes for POP one could mention the duration of labor, the parity, the birth weight of the neonatal, the menopause (as a factor of urogenital atrophy) and, also, combinations of anatomical, physiological, genetic and lifestyle reproductive factors [2]. Women with symptomatic disorders suffer physical and emotional distress which has negative impact on women's social, physical and psychological well-being.

Various laparoscopic and vaginal procedures involving uterosacral ligaments suspension have been developed, however surgery for POP is known to have a high reoperation rate. The aim of this study is to investigate, by means of an experimental animal model, the potentialities of Platelet Rich Plasma (PRP) for the treatment of POP.

## 2. Materials and methods

USLs are crucial for pelvic organ support and in case their stiffness is reduced they cannot properly undertake this role. USLs of women with POP are characterized by increased amounts of collagen III

and matrix metalloproteinase 2 (MMP-2). Collagen III is essential in tissue elasticity and extensibility, and it is, therefore, assumed that it is responsible for the tissue laxity. In this context, it was thought that USL could be enforced (and therefore restore the normal anatomy) by injecting an autologous adhesive factor. The use of autologous substances could limit the side effects of synthetic materials. Taking into account that PRP is extremely rich in growth factors and cytokines (which regulate tissue reconstruction) its use in treating POP appears worth to be assessed. PRP has been already used in various clinical fields (orthopedics, plastic surgery etc) but, to the authors' knowledge, not in urogynaecology.

In this context, an experimental protocol was designed and approved by the Veterinary Service of Attica Region, according to the 200/63/EU Directive. Forty two (42) female, adult, Wistar rats, were classified into four groups as follows:

- 14 rats were virgin (nulliparous),
- 13 rats had given birth once (primiparous),
- 8 rats had given birth two or more (multiparous),
- 7 rats were multiparous treated with PRP.

The rats were born in the facilities of the "Laboratory of Experimental Surgery & Surgical Research N.S. Christeas" of the National and Kapodistrian University of Athens. They were caged in separate cages in a controlled environment at 22° C with 12h light/dark cycles. Standard rat feed and water were provided ad libitum. PRP was prepared based on a standardized protocol and was injected in USLs, after surgical exposure of them. The rats were ready for study 6 weeks after injection.

Following sacrifice, the pelvic anatomy of the rats was defined by gross dissections to expose the vagina, the vaginal connective tissue attachments to the pelvic sidewall, the pubic symphysis, and the

levator ani muscles. The hind limbs were disarticulated at the acetabulum, and the spine was disarticulated above the L1 vertebra. All dissections were carried out according to the same way. The specimens were then placed in normal saline 0.9% and were submitted to direct tension within one hour, after sacrifice. An electromechanical MTS Insight frame was used, equipped with a calibrated load-cell of capacity equal to 100 N. For gripping the tissue a customized hook was placed in the uterus, through the posterior vaginal wall. The sacrum-uterus complex was placed horizontally. The force was exerted to the ligaments through the hook which was attached to the upper clamp. Another clamp fixed to the base of the frame was used to immobilize the sacrum (Fig.1). The tests were displacement controlled (0.4 mm/min) until fracture of one of the USLs. The force imposed and the tissues elongation were recorded permitting determination, also, of the tissue's stiffness.



Fig. 1. The gripping system improvised.

### 3. Results

The force-elongation plots had the typical form, expected for soft tissues, consisting of the toe region and an almost linear portion followed by a strongly non-linear one, leading to fracture. The plots for the group of multiparous rats treated with PRP are seen in Fig.1. The plots for the remaining three groups of rats are qualitatively similar.

The deviations for the maximum force,  $F_{\max}$ , attained were not dramatic for the rats of the same group, indicating satisfactory repeatability of the experiments. However, the values of  $F_{\max}$  did not exhibit statistically significant differences between the various groups of rats (obviously due to the dependence of the specific quantity on the absolute magnitude of the cross-section of each ligament, which shadows the actual differences).

On the other hand, analysis of the experimental

data revealed that the stiffness of the USLs of primiparous and untreated multiparous rats is decreased by more than 12% compared to the group of nulliparous rats (as it was, perhaps, expected taking into account that the ligaments of multiparous and primiparous rats are relaxed due to labor). Concerning the stiffness of the USLs of rats treated with PRP it was found to be increased by almost 22% in comparison to that of the group of untreated multiparous rats, offering, thus, a very strong indication that the treatment with PRP could be beneficial.

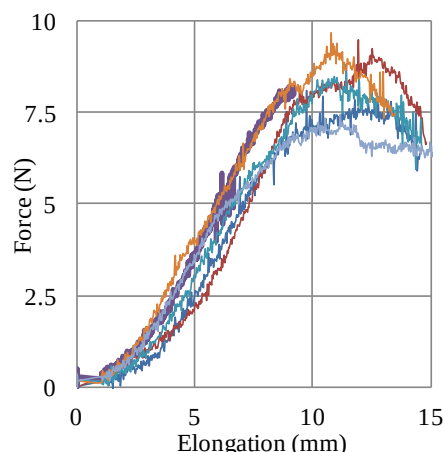


Fig. 2. Force-elongation plots for the group of multiparous rats treated with PRP

### 4. Discussion and conclusions

Although the research project described is still in progress, strong evidence was provided that the potentialities of PRP for non-surgical treatment of POP must be further explored. Limitations of the study that should be mentioned include: (i) The relatively small number of experiments per group that should be increased (it is anticipated, of course, that growing up rats in conditions, like the ones of the specific animal model, is extremely time consuming and there is a high financial cost for it). (ii) The use of non-normalized quantities (like force and elongation) instead of normalized ones (i.e., stresses and strains), which would relieve the results from their dependence on the size of each specimen. Finally, the surgery procedure should be standardized to avoid additional scattering effects.

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# BONE STRUCTURE INVESTIGATION IN A CASE OF AUTOSOMAL RECESSIVE OSTEOPETROSIS

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## 1. Introduction

Autosomal Recessive Osteopetrosis (ARO), also named malignant infantile osteopetrosis, is a group of disorders characterized by defective bone resorption by the osteoclasts, inducing a marked increase in bone mass and a high bone fragility. Hematopoietic Stem Cell Transplantation is the therapeutic option to correct ARO in patients with osteoclast-intrinsic disease and without neurodegenerative complications [1-3].

This paper presents the first extended description of the abnormalities still present in the ARO bone structure after correction by HSCT.

## 2. Material

Two subjects were investigated in this study: a ARO case that was corrected by HSCT at 6 months and a healthy control, whose characteristics and body mass index (BMI) are reported in Tab.1.

Table 1. Subjects description.

| Case                     | ARO                   | CONTROL         |
|--------------------------|-----------------------|-----------------|
| Diagnosis                | ARO corrected by HSCT | Healthy control |
| Gender                   | male                  | male            |
| Age (years)              | 25                    | 28              |
| BMI (kg/m <sup>2</sup> ) | 28                    | 21.4            |

## 3. Methods

The CT scans images of both subjects were acquired at IRCCS Burlo Garofolo using a Philips Brilliance scanner TC at KVP 120 kV, 130 mA, FOV 250 x 250, 1mm resolution, allowing the 3D reconstruction of the bone structure.

In one femoral head of each subject, a cubic volume of interest (VOI) with a side of 25 pixel and 25 slices per stack was selected. Although the VOIs consist of a small number of pixel, they were the largest cubes of equal size that could be extracted

from the reconstructions, given the resolution obtained in the acquisition. The same reconstructed trabecular volumes were used for the 3D morphological analyses and for the 3D numerical simulations.

Fiji, an open source project bundling a lot of plugins which facilitate scientific image processing, was used for selection, enhancement, normalization and OTSU segmentation [4].

As an example of the differences immediately visible in the two bone structures, Fig.1 depicts a cross-section of the cubic VOIs after segmentation, for the ARO (left) and the CONTROL (right).



Fig. 1. Cross-section of the segmented VOIs: ARO (left) and CONTROL (right).

### 3.1 3D morphological analyses

The morphological properties of the selected trabecular architectures were quantified and measured using the Quant3D software [5], which implements several commonly-used techniques such as the Mean-Intercept Length (MIL), Star Volume Distribution (SVD) and Star Length Distribution (SLD) methods [6] to compute:

- Bone Volume/Tissue Volume (BV/TV);
- Degree of Anisotropy associated with the z direction (DA\_M), ratio between the Mean Intercept Length eigenvalue in the axial direction and the average eigenvalue in the transversal plane;
- Trabecular Number (Tb.N);
- Trabecular Thickness (Tb.Th).

### 3.2 3D Cell Method simulations

The Cell Method is a numerical method based on a direct discrete formulation of equilibrium equations. Due to the direct discrete approach employed, no restriction is imposed by differentiability conditions and the characteristic length of the elementary cell of the discretization can be of the same order of magnitude as the heterogeneities of the structure. The Cell Method has been shown to be very efficient for the analysis of complex materials and porous structures, like trabecular bone. A Cell Method model, developed in for the simulation of compression tests along the coordinate axes of a trabecular bone volume was applied in this work [7-10]. An elastic modulus of 1000 MPa and a Poisson's coefficient of 0.3 were assumed for the bone/solid phase material.

The following parameters were calculated in the Cell method analyses:

- Elastic modulus  $E_z$ , along the axial direction;
- Average elastic modulus  $E_{xy}$  in the transverse plane;
- Degree of Anisotropy ( $DA\_E = E_z/E_{xy}$ ).

### 3.3 Bone Structure Index

The BESTEST<sup>®</sup> is a recently introduced analysis that can be used to quantify the quality of bone micro-architecture and its pathological alterations induced by age, pathological conditions or lack of exposure to physiological mechanical stimuli, by computing the Bone Structure Index (BSI) by Cell Method simulations on a virtual biopsy of the patient, acquired in the proximal phalanges of the non-dominant hand [11-15]. The BSI T-score, that is the number representing a person's bone quality compared with the average BSI value of young Caucasian women (at peak bone health), was evaluated in the two subjects.

## 4. Results

The parameters describing the trabecular structure in the two subjects are summarized in Tab.2.

## 5. Concluding remarks

This work offers an extended description of the abnormalities found in the bone structure of an ARO patient, that has never before been reported.

Our results clearly indicate that the trabecular structure of the ARO patient remains deeply altered at the morphological and structural levels even

several years following the hematopoietic stem cell transplantation, despite exhibiting a bone volume fraction that falls in the range of normal values.

**Table 2.** Results.

| Case            | Unit             | ARO  | CONTROL |
|-----------------|------------------|------|---------|
| BV/TV           | %                | 44.8 | 38.1    |
| DA_M            |                  | 0.44 | 1.17    |
| Tb.Th           | mm               | 2.53 | 1.35    |
| Tb.N            | mm <sup>-1</sup> | 0.18 | 0.28    |
| Ez              | MPa              | 60   | 190     |
| E <sub>xy</sub> | MPa              | 175  | 95      |
| DA_E            |                  | 0.33 | 2       |
| BSI T-score     | DS               | -2.1 | 1       |

## Acknowledgements

The Authors wish to thank V. Clemente and A. Nicolosi for their assistance during data collection.

Disclosure: FC is co-founder of M2TEST srl, the company that commercializes the BESTEST<sup>®</sup>.

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# DISPLACEMENT-ENCODED MRI REVEALS ALTERED STRAIN PATTERNS IN ENZYMATICALLY DEGRADED CARTILAGE DURING CYCLIC COMPRESSION

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## 1. Introduction

Osteoarthritis (OA) is a debilitating disease estimated to affect more than 40 million people across Europe and 1 in 10 of the population over 60 [1], and is characterized by the degeneration of articular cartilage. Cartilage is a connective tissue composed of an extracellular matrix consisting mainly of water, collagen, and proteoglycans, in which cartilage cells – chondrocytes – are embedded. With OA progression, the matrix constituents, particularly collagen fibrils and proteoglycans, become damaged, thus leading to a decrease in the tissue's mechanical properties.

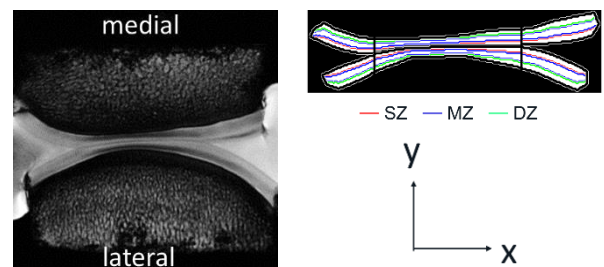
To achieve a better understanding of the role of mechanical loading in the onset and progression of degenerative joint disease such as OA, the contribution of the different cartilage matrix constituents to load bearing needs to be explored. The objective of this study was, therefore, to characterize and compare strains under compressive loading in intact, proteoglycan- and collagen-depleted cartilage plugs, using an experimental setting simulating *in vivo* cartilage-on-cartilage contact conditions.

## 2. Materials and methodology

Bovine knee joints from 9-month-old calves were obtained from a local abattoir within one day of slaughtering. Cylindrical osteochondral plug pairs with a diameter of 30 mm were harvested from the load-bearing area of the knees, one from the medial and one from the lateral femoral condyle. After preparation, plug pairs were randomly assigned to one of the following groups: one group (n = 4) was treated with 6U/ml collagenase type VII (Sigma Aldrich, UK) to induce collagen

degradation [2]; another group (n = 4) was treated with 0.1 U/ml chondroitinase ABC (Sigma Aldrich, UK) to induce proteoglycan (PG) degradation [3]; a third (n = 4) was treated with a combination of the two enzymes to induce both PG and collagen degradation; the last group (n = 3) was kept in DMEM:F12 medium and served as controls.

The paired plugs were then mounted in a custom-built loading device [4], with the lateral plug fixed in the moving part while the medial was static, and cyclically loaded in axial compression within a 9.4T MRI scanner (Biospec94/20 USR, Bruker Biospin, Ettlingen, Germany). Each cycle consisted of loading up to 350N, 2.0s holding time, and unloading in 1.4s. First, the explants were loaded for approximately 500 cycles to reach quasi-steady state load-deformation behavior. Then, image acquisition during compressive loading was performed using a DENSE sequence (Displacement Encoding with Stimulated Echos), which allows for measuring displacements in the cartilage from the MRI phase images [5].



**Fig. 1.** Left: MRI DENSE reference scan; Right: Segmented cartilage layers, zones, contact area, and displacement directions

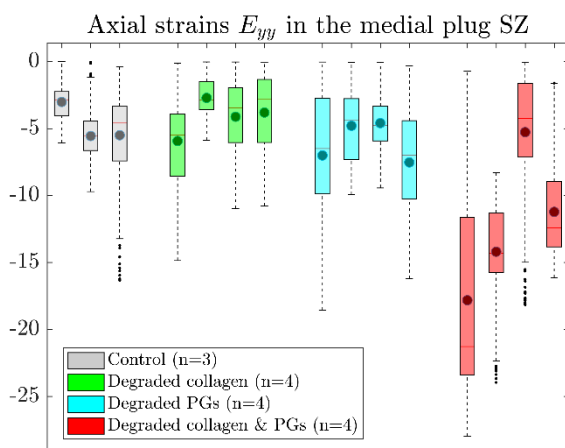
The raw data was processed using MATLAB (2015a, The Mathworks, Natick, MA) and phase differences between reference and displacement-

encoded scans (size: 512x512 pixel<sup>2</sup>, resolution: 0.1172 mm) were converted into transverse (dX) and axial displacements (dY). Green-Lagrange transverse ( $E_{xx}$ ), axial ( $E_{yy}$ ), and shear strain components ( $E_{xy}$ ) were calculated pixel-wise using a maximum likelihood estimation of the deformation gradient tensor. Strain comparison between intact and degraded sample pairs was performed in the contact area between the cartilage layers, for the superficial (SZ), the middle (MZ), and the deep zone (DZ), shown in Fig. 1.

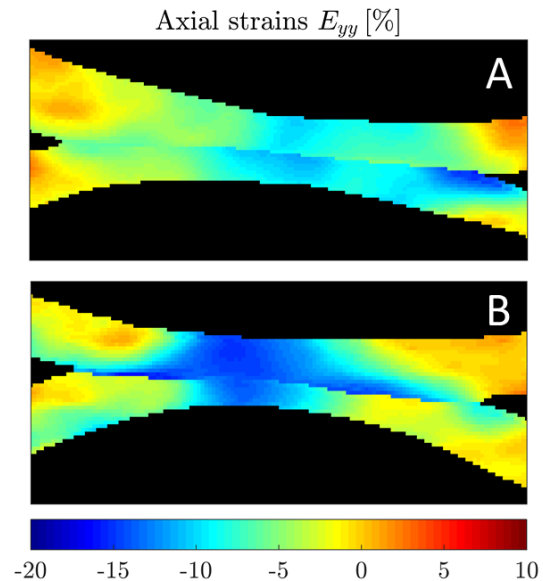
### 3. Results

The contact-area specific transverse, axial, and shear strains were higher in the degraded samples than in the control samples, with the samples degraded with combined collagenase and chondroitinase ABC showing the largest strain deviations from controls in all three directions. Interestingly, the spread of the strain distributions in the three zones of the cartilage plugs was larger in the degraded than in the control samples, indicating that enzymatic degradation induced a higher material heterogeneity (Fig. 2).

The most dramatic differences between control and combination-degraded samples were noticeable in the superficial zone (SZ) of the medial (static) plug, in the axial (y) direction (Fig. 2 and Fig. 3). The reason for this effect is twofold: i. due to the limited penetration of the enzyme in the cartilage tissue, the SZ is most affected by the enzymatic degradation; and ii. the axial direction coincides with the direction of compression.



**Fig. 2.** Pixel-specific strain values in the contact area in the superficial zone (SZ) for each sample



**Fig. 3.** Axial strain map in a control (A) and a combination-degraded (B) sample

### 4. Conclusions

Degradation of the main components of cartilage, collagen and proteoglycans, changes the biomechanical response of the tissue during cyclic compressive loading. Nevertheless, while degradation of one or the other matrix component induced low strain increase, samples depleted of both collagen and proteoglycans showed significantly higher maximum axial, transverse and shear strain components than controls. This indicates that the interplay between cartilage matrix constituents plays an essential role in the decrease of the load bearing capacity and the altered mechanical environment characteristic of degenerative joint diseases such as OA.

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# NONLINEAR DYNAMICS OF HUMAN KNEE JOINT DURING JUMPS

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## 1. Introduction

In everyday life there are situations when jumps over obstacles are necessary during normal walking. Such cases lead to stability problems in the lower limb joints during the flight phase and landing. Common clinical applications like identifying movement disorders, assessing surgical outcomes, improving walking stability, and reducing joint loading need gait monitoring for understanding the nonlinear gait pattern. There are numerous papers in the scientific literature that deal with walking stability using nonlinear dynamic analysis methods [1-6], but very few papers treat joint stability during the three phases of jumping: takeoff phase, flight phase, and landing phase. In bilateral foot landing the knee acts as the primary shock absorber. So, developing a dynamical model for the human jumping process study leads to the evaluation of the effect of factors like joint speeds in knee stability knowing that landing applies a significant impact load on the limited contact area of the knee components, thereby causing great stress and increasing the risk of injury.

In this paper we aim to analyze the human knee joint in frontal plane with non-linear dynamic analysis instruments such as Lyapunov exponents in order to quantify the local stability during jumps.

## 2. Experimental test

The experimental method suppose the use of a Biometrics data acquisition system based on wearable electrogoniometers and Data Log acquisition device [1]. The block schema of the acquisition data system is presented in Fig. 1. Six trials of jumping test were performed by a healthy male subject. The angular amplitudes of human knee rotation in frontal plane during the jump were obtained for each trial as data files.

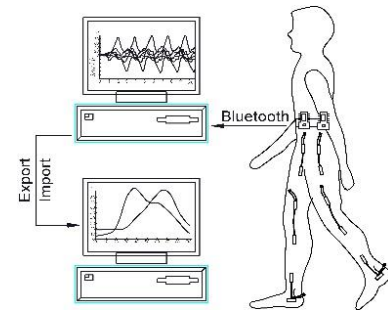


Fig.1.

The angle defined in the frontal plane is shown in Fig. 2.

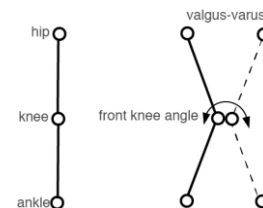


Fig. 2. Angular displacement of the knee joint in frontal plane

The evolution of the knee joint in frontal plane for one trial is depicted in Fig. 3. The number of samples is the number of points in the time series.

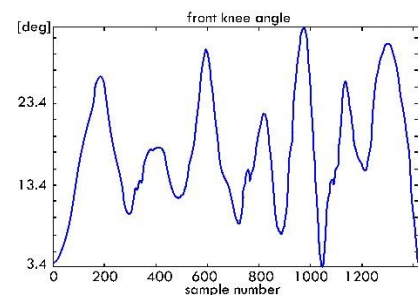


Fig. 3. Angle of the knee joint in frontal plane

The system represented in the three dimensional space  $[s(n), s(n+\tau_0), s(n+2\tau_0)]$  is shown in Fig. 4. The samples of the goniometer system are  $s(n)$ .

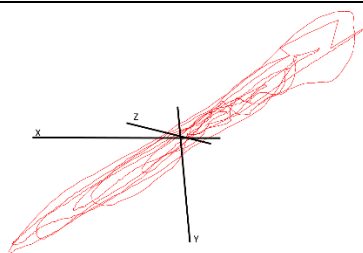


Fig. 4. Attractor in three dimensional space

For the reconstruction of the phase space the time delay is calculated as depicted in Fig. 5. Time delay for the trial in Fig.5 is  $T=12$  and was calculated using the average mutual information [3, 4].

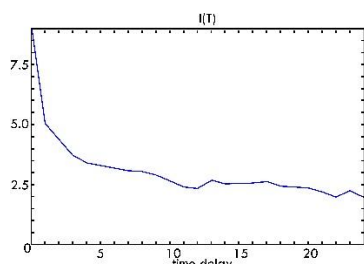


Fig. 5. Time delay

The embedding dimension  $d_E$  is determined using the false nearest-neighbor (FNN) method [7]. The total percentage of false neighbors declines and  $d_E$  is chosen being  $d_E = 3$ , where this percentage approaches zero [4]. The Lyapunov exponents,  $\lambda_i$ , provide a measure of the sensitivity of the system to its initial conditions. In order to determine the behavior of a dynamical system the largest Lyapunov exponents (LLE) must be calculated. The value of LLE quantifies the exponential divergence of the neighboring trajectories in the reconstructed state space and reflects the degree of chaos in the system. The average logarithmic divergence function of time is shown in Fig. 6. The slope of the logarithmic relation represents LLE [2, 8] which are calculated using the Rosenstein method [8].

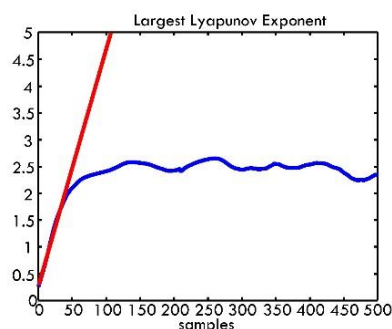


Fig. 6. LLE represents the slope of the logarithmic relation

Figure 7 shows the LLE for the knee joint for all the six trials. The values are comprised in the interval  $[0.034; 0.047]$ .

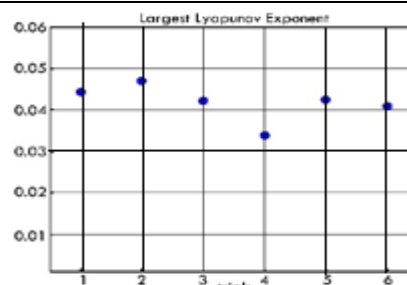


Fig. 7. Largest Lyapunov exponents

### 3. Conclusions

The purpose of this study was to investigate the biomechanics of chaotic characteristics of rotation movements in frontal plane of the knee joint during the jumps. We applied the chaotic analysis to the data series of the angular amplitudes of human knee rotation in frontal plane collected by using Biometrics system based on twin-electrogoniometers. The LLE obtained for each trial of human knee joint were positive values which suggest that human knee motions show chaotic characteristics. Larger values of LLEs are associated with more divergence, a decrease of local stability and increase of knee rotation variability.

The Lyapunov exponents can be used to characterize the human jumps activity.

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# PERFORMANCE MEASUREMENT OF SAILORS WITH CUSTOM HIKING BENCH AND MOTION TRACKING SYSTEM

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## 1. Introduction

The development of professional athletes is based on the measurement and analysis of their performance. The most determinant event during the sport should be defined [1]. The implementation method of that event is critical in the perspective of the sport performance. One of these key events during sailing is the so called hiking, when the athlete's posture is almost horizontal on the boat, trying to keep the boat upright against the forces of the wind.

In the case of sailing, due to fundamental reasons (e.g. we need a big area on the water), the performance can be measured only in artificial conditions, where the simulation of the boat is essential. The known methods for examining the hiking posture are: recording the physiological changes during hiking (pulse, gas exchange) [2]; measurement of the forces, torques and joint angles during hiking and comparing the results with the maximal performance [3]; realistic simulation of sailing with graphic display and playable software [4].

The present research aims to develop a measuring method for analyzing the hiking technique of sailors. For that, a custom hiking bench was built and equipped with measuring devices to record the exerted forces and the hiking posture of the sailor.

## 2. Methods

### 2.1 Design and implementation of the hiking bench and the measuring equipment

A static hiking bench, built after our custom design, was used as a boat simulator. The two-sided structure's dimensions can be adjusted to the size of a Laser and to the size of a Finn dinghy hull (Fig. 1.).

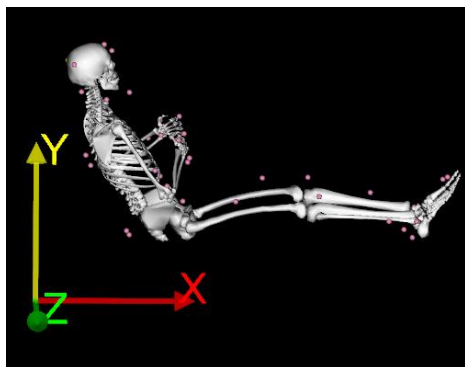


**Fig. 1.** The custom hiking bench with the subject on it

During the measurement the subject is in hiking position on the bench for a predetermined time interval. Meanwhile, the forces on the hiking strap (the sailors are holding on to the middle of the boat with their feet hooked in this strap) and on the simulated sails are recorded. A fitness machine's masses were used to simulate the forces of the wind on the sail, which are connected to the hiking bench via the so called schott rope. These forces are recorded with modified hanging scales supplemented with a HX711 ADC and an Arduino® Mega 2560 microcontroller. The measurement is coordinated by an NI LabVIEW™ 2016 (National Instruments™, Austin, Texas, U.S.) algorithm (developed in our lab).

Beside the forces on the boat, the posture of the sailor is recorded with means of reflective markers, fixed on the subject's clothing, and the OptiTrack® Motion Capture camera system (NaturalPoint, Inc. DBA OptiTrack), consisting of 18 calibrated cameras (Flex 13). The cameras record the 3D-coordinates of the markers, which are documented by the original Motive® (NaturalPoint, Inc. DBA OptiTrack) software and processed with the OpenSim software (Fig. 2.) to get the joint angles and the CoM (Center of Mass) coordinates. The exported joint angles are: knee bending-, ankle bending-, hip tilting-, hip bending-, hip rotation-,

lumbar tilting-, lumbar bending- and lumbar rotation angles.



**Fig. 2.** The subject's implementation in the OpenSim software

## 2.2 Subject and experimental protocol

The subject involved in the study was a professional Finn dinghy sailor (age: 15 years; height: 193.2 cm; body mass: 92.1 kg). Prior to the measurement, the hiking bench was set to the dimensions of a Finn dinghy boat and the sailor did a one-minute warm-up hiking on the bench.

The first part of the measurement consisted of three 180 sec hiking sessions with ~10, 15 and 20 kg loads on the schott rope, which simulated 6-10, 10-12 and 12-16 knots wind speeds [3]. The subject was asked to make movements according to the simulated wind speed of the sessions. In the second part, the maximal physical performance was measured by recording the maximal force exerted by the arms and a Bucket test. During the Bucket test the subject had to perform a static hiking position on the hiking bench, while holding the schott rope with both hands symmetrically. On the other end of the rope, the load was increased by 5 kg/30 sec starting from 10 kg. During the hiking sessions and the Bucket-test, the forces on the hiking strap and on the schott rope were recorded along with the posture of the sailor.

## 3. Results

The results of the hiking sessions were compared to past measurements carried out on water summarized in [3]. The profiles of the forces on the hiking strap were similar, but the profiles of the forces on the schott rope were not comparable. The hiking postures, by which the greatest and the least forces were exerted on the hiking strap, were compared separately at the three simulated wind speeds. Depending on the schott load, the CoM coordinates did not change significantly, while the hiking posture (the joint angles) did vary

significantly. The hiking strap forces recorded in the 6-10 and 12-16 knots simulated wind speeds were the greatest when the subject leaned back in the right direction and turned in a counterclockwise direction, which complies with the movement carried out on water as a reaction to waves. The results of the hiking sessions and the maximal physical performance tests were also compared. During the Bucket test, significant changes show in the hiking posture when the subject reached his maximal capacity. The maximal force on the schott rope was the 91.77% of the maximal force exerted by the arms and the maximal force on the hiking strap was 125.08% of the subject's body mass, which shows that the sailor was in excellent physical condition.

## 4. Conclusions

The designed hiking bench and measuring equipment is suitable for measuring and evaluating sailors' hiking performance. It can be stated that the hiking technique depends on the wind speed and this can be described by numerical data. The sailors and their trainer can get feedback on the applied hiking technique through this performance measurement. The trainer can give specific advices to his sailors for achieving better results.

To develop the hiking technique comprehensively, more sailors from different levels should be measured and compared in the future.

## Acknowledgements

This work was supported by the Hungarian Scientific Research Fund OTKA [grant number 115894].

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# ANALYSING HUMAN BALANCE RECOVERY ACTION USING CALCULATED TORQUES OF A DOUBLE PENDULUM MODEL

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## 1. Introduction

In clinical practice, assessment of balancing abilities aids the diagnostics of various fields such as orthopedics and neurology while researching balance can provide insight for control engineering. One test method uses a freely oscillating platform with a sudden perturbation to elicit balance recovery response motion [1]. Standing balance is recovered and maintained using dominantly the ankle or a mixed hip-ankle strategy for bipedal (BP) and unipedal (UP) stances, resp. In this proof of concept work, we demonstrate the utility of a 4 degrees-of-freedom (DoF), two-link double pendulum (DP) model of the human body according to the Denavit-Hartenberg convention [2] in calculating the estimated forward and sideways torques at the ankle and hip level based on actual position data from a motion capture system.

## 2. Materials and methodology

### 2.1 Participants and balance measurement

Eleven young, healthy collegiate participants (9M/2F; age:  $22.8 \pm 1.7$  yrs, height:  $179 \pm 11$  cm, body mass:  $80.6 \pm 19.9$  kg) were tested 10 times in three stances each: BP, right UP, left UP stance. The tests were authorized by the Science and Research Ethics Committee of Semmelweis University (174/2005) and written consent was obtained from all participants. Perturbation balance tests (Fig. 1) were performed on the PosturoMed® freely oscillating platform (Haider Bioswing, Weiden, Germany) as previously described in [1]. Briefly, the suspended platform is locked outside its resting position with the participant adopting a UP or BP stance; a sudden release delivers a  $\sim 20$  mm unidirectional perturbation. The task is to regain balance via stopping the oscillatory motion of the platform. Other than the standing support surface, no objects could be touched; foot contact with the platform had to be constant and rigid, and platform oscillation had to be decreased below  $\pm 2$  mm.

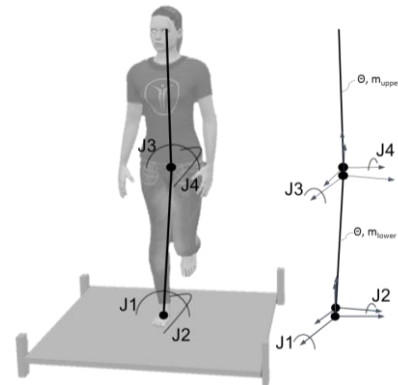


Fig. 1. Double pendulum model of balancing person on unstable platform.

### 2.2 Double pendulum model

The DP model (Fig. 1) consists of a lower body link (feet, shanks, thighs) and an upper body link (pelvis, trunk, arms, neck and head) with four single rotational DoF joints. Feet and arms were modeled as mass points; left and right shanks and thighs as two-two cylinders each; trunk-pelvis and neck-head as one-one cylinders each. Weight, inertia, center of mass values were based on proportional values [3] of total body mass and height. The respective body segments for the two DP links were then summed up to obtain the final inertial properties. Dynamics of the DP model were obtained through the Denavit-Hartenberg convention [2]. Joint1 and Joint2 allows lateral and frontal rotation in the ankle, resp.; Joint3 and Joint4 allows lateral and frontal rotation in the hips, resp. (Fig. 1). The four equations of motion give the calculated torque for each joint.

### 2.3 Data collection and analysis

Position data of platform, hip and head were collected using an 18 infra-red camera motion capture system OPTITRACK (NaturalPoint Inc., Oregon, USA) with passive reflecting marker sets at a sampling rate of 120Hz. Joint angles were calculated from position data. Angular velocity and acceleration were obtained by differentiation and values were low-pass filtered. Evaluating the equations of motion for these gave the joint torques.

Total joint work was calculated integrating torque multiplied by angular velocity.

### 3. Results

In order to provide a proof of concept of the model, the derived variables are explored to see if they can indicate the ankle or mixed strategies and if they are congruent with platform trajectory. Out of 330 attempts, 285 was successful and well recorded for the whole group. Here, results of a single participant are shown as an example.

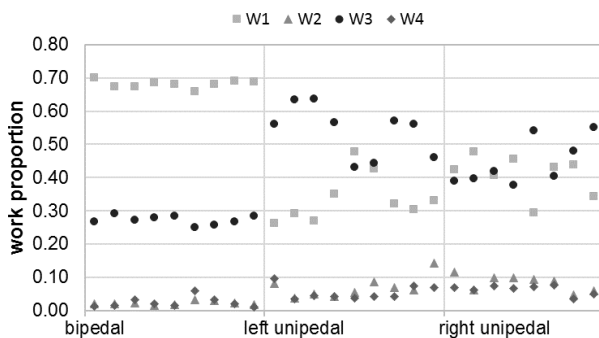


Fig. 3. Total joint work proportions of recovery motion.

Total work in the four joints can be analyzed in relative terms to identify if one or more is dominant (Fig. 3). The BP stance very consistently shows dominant and lower lateral torques at ankle (W1) and hip (W3) level, resp., with minimal frontal torques (Fig. 3). The dominance of lateral ankle torque (W1) over hip torque (W3) reverses for UP stances (Fig. 3). Left and right UP stances are also somewhat dissimilar in this regard. Frontal torques (W2, W4) also appear for UP stances (Fig. 3).

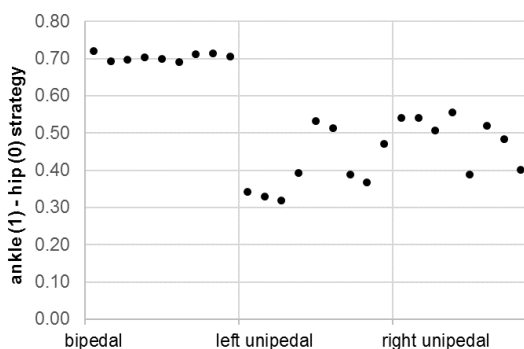


Fig. 4. Indication of ankle (1) and hip (0) strategies, based on joint work proportions.

Summation of ankle joint works (W1, W2) indicates joint dominance on a simple 0-1 (hip-ankle) scale; BP and UP stances are clearly distinct (Fig. 4). A lower directional ratio indicates more elliptical platform motion in response to the purely lateral perturbation, thus utilizing forward-backward motion to recover balance. The ratio of

lateral (Q1, Q3) over frontal (Q2, Q4) joint works correspond well with platform motion (Fig. 5). For the total 11 participants, observations on the relative proportion of joint works were fairly consistent. However, the seemingly linear trend as depicted in Fig. 5 was similar only for 5 participants.

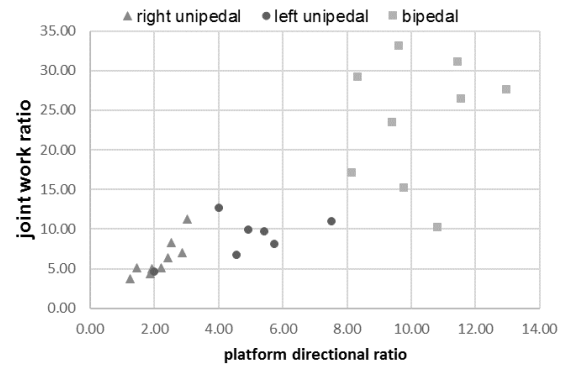


Fig. 5. More circular platform motion is elicited by antero-posterior torques.

Overall, the model allows to further analyze and better understand balance maintenance and the complex balance recovery action. In the future, the model could be evaluated for different types of balancing tests used in clinical practice.

### 4. Concluding remarks

- Measured position data were input to a double pendulum model to calculate joint torques.
- The model indicates dominance of ankle or hip joint strategies in different balance recovery conditions.
- Calculated torques relate to the trajectory shape of recovery motion.

### Acknowledgements

The assistance of Simon Herlett and Bence Fulop with software coding and consultations with Ambrus Zelei, Ph.D are acknowledged. This project was supported by Hungarian Scientific Fund K115894. Balint Petro was supported by Erasmus Mundus PANTHER Project grant during this work.

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# INVESTIGATION OF THE DESIGN ASPECTS OF RUNNING SUITS APPLYING MOTION ANALYSIS

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## 1. Introduction

Compression garments (CG) has been used in medicine for many years. Because of the external compression the linear velocity of the blood flow increases inducing an improved blood circulation [1]. For this reason, its widely used for therapeutic and prevention purposes, e.g.: prophylaxis against lymphoedema or thrombosis [2]. In terms of their efficiency in exercises, they can reduce oscillation and increase strength enhancing the performance. According to MacRae et al. the level of pressurization is a significant variable in compression garments [2].

The goal of present research is to compare a purchased compression sportswear with the same cut, but normally dress editing garment applying motion analysis. Based on the motion analysis we can assume some remarks in connection with the design of the sportswear.

## 2. Method

### 2.1 Investigated running suits

During the investigation two different sportswear were compared. Firstly, an unprofessional compression running tight (Nike, Power Speed Tight) was tested which has a 4-5 mmHg compression on the thighs, 6-7 mmHg on the knee region and 13-16 mmHg on the calves. It is composed of 74% polyester and 26% elastane with 100% polyester gusset lines (Fig. 1 a.). Additionally, an elastic long sleeve top (Asics, Inner Muscle LS Top) was worn which made of 84% polyamide and 16% elastane. On the other hand, a same cut but normally dress editing suit was tested. The set of garments were standardized with a 1% undercut, so there was no effective grip or compression on the body and it composed of 74%

polyamide and 26% spandex. Both the tight and the top were designed and sewn individually to the females involved in the experimentation (Fig. 1 b.).

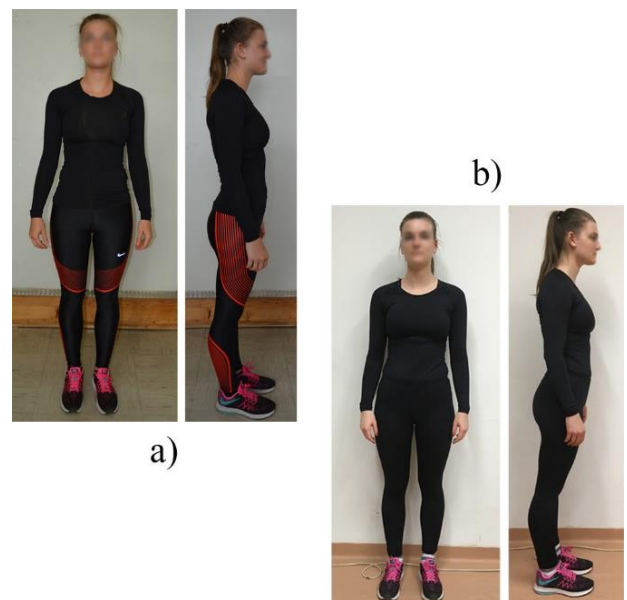


Fig. 1. a) compression sportswear, b) self-designed sportswear

### 2.2 Subjects

Four healthy, well-trained female free time runners (age  $21.0 \pm 2.2$  yrs, body mass  $63.3 \pm 2.1$  kg, height  $170.5 \pm 2.5$  cm, BMI  $22.6 \pm 1.7$  kg/m<sup>2</sup>) participated in the study which was approved by the Hungarian National Science and Research Ethics Committee (21/2015). None of the participants had suffered from any injury which affects to the locomotion system and they arrived at the measurement in sufficiently relaxed, well-hydrated and well-nourished condition.

### 2.3 Measurement procedure

During the measurements an OptiTrack (NaturalPoint, Corvallis, Oregon, USA) based optical motion analysis system was used with 18

pieces of Flex13 type tracking cameras and the Motive:Body motion capture software. Prior the measurement, all the subjects were instrumented with 39 retroreflective markers applying a convention full body markerset. The experimentation was performed on a treadmill applying gradually increasing velocity intervals. After 3 minutes walking the following velocities were considered: 7 km/h, 8 km/h, 10 km/h, 11 km/h. The participants ran 3 minutes in each period without any break.

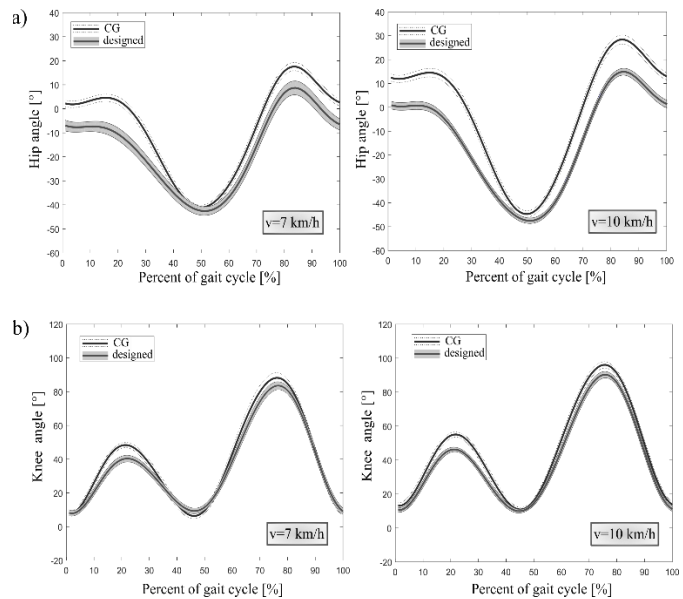
## 2.4 Data Analysis

Applying the 3D position data of the markers the angular and geometrical parameters were calculated using an open-source software OpenSim (NIH Center for Biomedical Computation, Stanford University, USA) [3]. Afterwards, a self-developed Matlab (R2017a, The MathWorks, Massachusetts, USA) script was utilized to split the motion into gait cycles and determine the relevant gait parameters. In this study the range of motion of knee extension (knee ROM), the range of motion of hip extension (hip ROM) the center of mass (COM) and the step length are considered. For all the parameters a mean value and its standard deviation were calculated and from these the coefficient of variation (CV) was also determined. Considering the difference between the running suits a simple one-sample t-test was performed with significance level of 0.05.

## 3. Results and Discussion

Comparing the kinematic parameters, it can be concluded that a larger deviation occurred in the hip angles and the trajectory of COM in y direction in the lower velocities in case of the self-designed sportswear. On the other hand, in higher velocities (10-11 km/h) the same deviation could be obtained (Fig. 2). The reason might be the more transpired moisture during the exercise. Note, that the cause of the high change in deviation in case of the self-designed garment should be investigated in more detailed.

Therefore, during the statistical analysis only those parameters were considered which was measured in the higher (10 km/h and 11 km/h) velocity intervals. The results shown that the hip ROM changed significantly in case of compression tight which may have occurred by the gusset lines in that region. Additionally, there is a significant difference in the more pressurized knee region in the knee ROM parameter.



**Fig. 2.** Measured **a)** hip angle values and **b)** knee angle values as a function of the gait cycle measured in 7 and 10 km/h in both sportswear

## 4. Summary

The aim of this study is to investigate the design aspects of a purchased and a self-designed sportswear. Applying motion analysis some improvements could be assumed. The more pressurized region results a significant difference in the gait parameters. Despite of the individually-made dress editing using a 1 % undercut is not as proper as the purchased garment.

## Acknowledgements

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## HUMAN BALANCING IN THE SAGITTAL PLANE

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### 1. Introduction

Examination of human balancing is becoming more and more important nowadays since many accidents can be connected to falls caused by the loss of balance, especially among the elderly. Moreover, theoretical and experimental analysis of human balancing helps to understand the control concept employed by the nervous system.

Human balancing is the stabilization of the human body around an unstable equilibrium and is controlled by the brain. Due to the particularities of our neural system, the feedback mechanism requires time, therefore the control concept has to be modeled with a delayed feedback. In this study we assume that the brain controls by signals proportional to the angular displacement and velocity of the human body and the balance board relative to the environment, which corresponds to a delayed PD controller, where the constant time delay  $\tau$  is the reaction time of the balancing subject.

### 2. Mechanical model

In this work, we analyze balancing on a rolling adjustable balance board in the sagittal plane. The balance board was manufactured at the Department of Applied Mechanics from plywood. The radius  $R$  of the arcs and the distance  $h$  of the board measured from the center of the arc can be changed, as it is shown in Fig. 1 and 2.

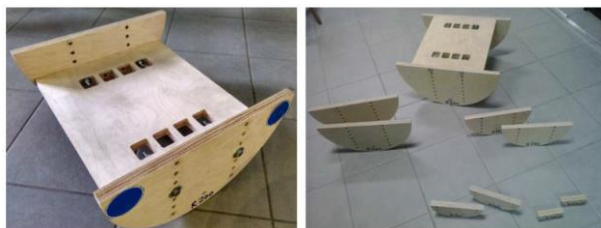


Fig. 1. Adjustable geometry of the balance board.

The two-degree-of-freedom mechanical model (Fig. 2) consists of the balance board and the balancing subject, which is modeled as a homogenous rigid bar. The feet are considered to be the part of the balance board. The ankle joint (characterized by parameter  $h_a$ ) is modeled by a torsional stiffness  $s_a$  [1]. Corresponding to the balancing trials, the joint is placed on the left side of the symmetry axis of the balance board.

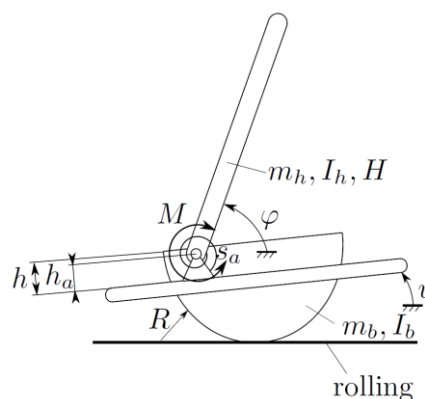


Fig. 2. Mechanical model.

The control torque

$$M(t) = P_\varphi \varphi(t - \tau) + D_\varphi \dot{\varphi}(t - \tau) + P_\vartheta \vartheta(t - \tau) + D_\vartheta \dot{\vartheta}(t - \tau) \quad (1)$$

is applied at the ankle joint, where  $P_\varphi$ ,  $D_\varphi$ ,  $P_\vartheta$  and  $D_\vartheta$  are the proportional and derivative control gains of the balancing subject and the balance board angle, respectively.

The governing equations reads

$$\mathbf{M}\ddot{\mathbf{q}}(t) + \mathbf{S}\mathbf{q}(t) = \mathbf{P}\mathbf{q}(t - \tau) + \mathbf{D}\dot{\mathbf{q}}(t - \tau), \quad (2)$$

where  $\mathbf{M}$  is the mass,  $\mathbf{S}$  is the stiffness matrix,  $\mathbf{P}$  and  $\mathbf{D}$  stand for the matrix of proportional and derivative control gains, respectively.

The vector of generalized coordinates is

$$\mathbf{q}(t) = \begin{bmatrix} \varphi(t) \\ \vartheta(t) \end{bmatrix}, \quad (3)$$

where  $\varphi$  is the angle of the balancing subject, and  $\vartheta$  is the angle of the balance board (see Fig. 2).

It is assumed that there exists a critical time delay  $\tau_{crit}$  for each combination of  $R$  and  $h$ . For delays greater than the critical there exist no control gains associated with a stable equilibrium.

### 3. Stabilizability of the system

The goal of the stability analysis is to determine the critical feedback delay for the adjustable  $R$ - $h$  pairs with semidiscretization [2] and other numerical techniques. The representation of the critical time delay over the  $R$ - $h$  plane gives the so-called stabilizability diagram, which is shown in Fig. 3.

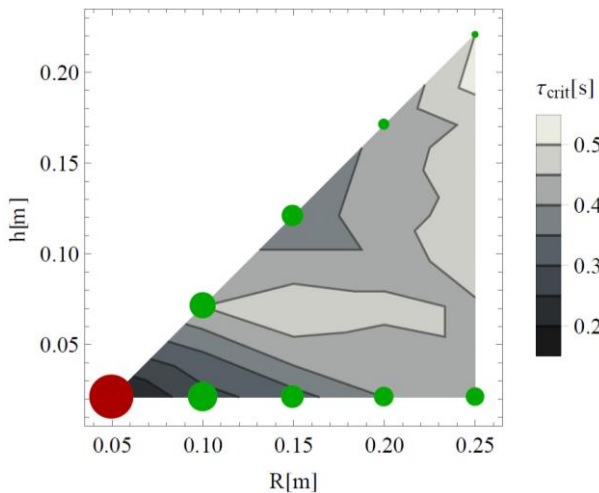


Fig. 3. Stabilizability diagram and measurement results.

### 4. Measurement

In order to compare the numerical stabilizability diagram with balancing trials, measurements were performed with volunteers in a systematic way. With the help of OptiTrack motion capture system and reflective markers (denoted by red in Fig. 4) placed on the balancing subject and the balance board, the “difficulty” of the balancing task can be analyzed with respect to  $R$  and  $h$ . The larger the standard deviation of the balance board angle, the “more difficult” the balancing task. The goal of the measurement was to determine the stabilizability boundary with respect to  $R$  and  $h$ .

The task of the subjects was to balance for 60 s with stretched legs and opened eyes on the balance board, as shown in Fig. 4. Green circles denote the

stable  $R$ - $h$  pairs in Fig. 4, where the diameter of the circle is proportional to the deviation of balance board angle and red color refers to the unstable trials.



Fig. 4. Measurement setup.

### 5. Conclusion

The reaction time of the balancing subject can be estimated after comparing the contour lines of the stabilizability diagram and the measurement results, which is approx. 0.25 s. The change of the deviation and the contour lines show the same tendency. In case of smaller  $R$  and  $h$  parameters, the critical time delay decreases and the deviation increases which implies more unstable balancing.

The reaction time of the balancing subject was measured in a simple way (pushing pedals as response to 10 light signals provided by a special device) before the balancing trials. The result was approx. 300 ms, which is greater than the delay obtained from the theoretical approach. This might indicate that in case of balancing we use not only visual but vestibular feedback.

In the future, the mechanical model will be extended with the angle of the trunk as an additional degree of freedom, and the dead-zone of the controller will also be considered.

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# 3D-PRINTED ANKLE-FOOT ORTHOSIS: A DESIGN METHOD

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## 1. Introduction

Additive manufacturing technology offers several advantages when compared to conventional manufacturing techniques, especially where large scale customizing is requested. This is particularly common for certain medical applications such as orthopedics, in which the efficiency of a treatment is strongly connected with each individual patient's body shape. [1]

An Ankle-Foot Orthosis (AFO) is an external medical device which helps a patient hold his/her ankle in the right position when the muscular/skeletal system is not sufficient alone (Fig. 1). The case study concerns a 21-year-old woman who had difficulties in walking after wearing a plaster cast for three weeks; an AFO would help her post-traumatic rehabilitation.

The more the shape of the orthosis adapts to the patient's body, the more comfortable and efficient the treatment will be. This paper describes a method to design a fully-customized Ankle-Foot Orthosis, taking advantage of 3D-scanning and 3D-printing techniques.



Fig. 1. An example of a standard Ankle Foot Orthosis.

## 2. Materials and methodology

The adopted design process can be divided into the following steps.

### 2.1 Foot scan with photogrammetry

The first phase is the patient's data acquisition; in this case the exact geometry of the ankle is required. One possible way of obtaining this data without the use of cumbersome and expensive 3D scanners is photogrammetry [2]. 150 photos of the patient's foot were taken from different angles and were then loaded into the Agisoft PhotoScan Pro™

software. PhotoScan™ elaborates a 3D point cloud of the ankle, which is then converted in a 3D mesh of the part, that can be exported in .stl format at the desired resolution.

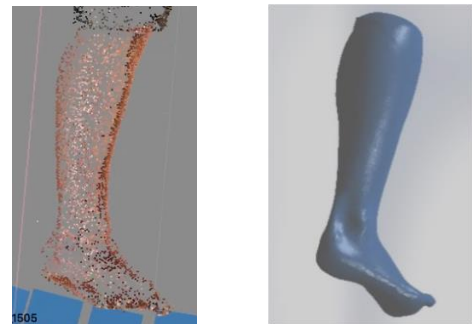


Fig. 2. From point cloud to mesh in Photoscan™.

### 2.2 Data import in CAD modeler

Having obtained a detailed mesh of the ankle it is necessary to import it into a CAD modeler in order to design the AFO around the scanned foot. SolidWorks™ Premium Edition 2014 CAD software was chosen for this purpose.

Direct editing of large .stl (or equivalent mesh files) in SolidWorks™ is very complex, if not nearly impossible, due to the numerous, inevitable, self-intersecting faces which often occur during the meshing process. Thanks to the *Scan to 3D* add-in, the program manages to fix most errors automatically, however, in order to perform any modelling around the imported object, SolidWorks™ must recognize it as a *surface* or *solid body*. In most cases this process is done automatically, but for complex geometries such as anatomical shapes the process is more elaborate.

Using the *Curve Wizard* and the *loft* features, it is possible to manually retrace the shape of the scanned foot, thus obtaining a fully-functional *solid body* around which it is possible to proceed with the AFO design.

### 2.3 AFO modelling

The *Offset surface* and *Thicken* features in SolidWorks™ allow the creation of a thin layer of material around the foot model. It is advisable to

leave 1-2 mm of free space between the skin and the inner surface of the orthosis to compensate for dimensional inaccuracies and to ensure greater comfort. This will be the starting point for the orthosis design. By performing an *extruded cut* with the desired shape, we obtain a first draft model of the device.

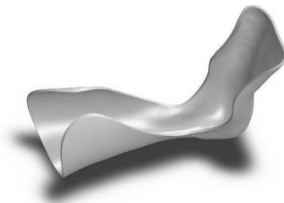


Fig. 3. A first draft of the AFO model.

#### 2.4 FEM optimization and 3D print

Before 3D-printing the modelled AFO, it is necessary to verify its thickness and outer shape in order to meet structural requirements. Given the medical function of this orthosis, a verification of the device under the loads and constraints described in Table 1 is sufficient.

Table 1. Loads and constraints applied to the AFO

|                    | Load                   | Constraint     |
|--------------------|------------------------|----------------|
| Type and magnitude | Vertical force<br>50 N | Fixed geometry |
| Application area   | Foot instep            | Calf           |

Fused Deposition Modelling (FDM) has been chosen for the print: this technology is currently the most widespread and inexpensive, still ensuring good overall performance, both in terms of mechanical resistance and of dimensional accuracy. The material chosen for the print is pure PLA, whose mechanical characteristics after 3D printing have already been tested [3].

FEM structural analyses were performed with SolidWorks™ *Simulation*. FDM 3D-printed materials exhibit a strong anisotropy, which must be considered when performing the stress analyses. Any custom anisotropic material can be defined in SolidWorks™ by manually inserting the Young's modulus and maximum tensile strengths in each direction. After performing each analysis, the profile can be adjusted to guarantee functionality in any problematic area.

Once the orthosis is modelled and optimized, it can be 3D-printed. In order to experimentally validate the whole process, the AFO should always be tested on the patient, under the supervision of a physiotherapist.

### 3. Results

The photogrammetric acquisition was able to give very detailed results. Much attention was put in maintaining dimensional accuracy during the CAD import in SolidWorks™: thorough inspections show excellent correspondence between the original photogrammetric mesh and the lofted body. FEM preliminary analyses show that the area that undergoes the greatest stress is around the heel. Therefore, thickening of the profile is required in this area (Fig. 4).

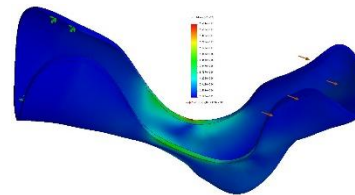


Fig. 4. FEM analysis: Von Mises equivalent stress.

### 4. Conclusions

The advantages of a custom-designed 3D-printed orthosis on a standard-type or hand-crafted device concern patient's comfort and ease of automation for the manufacturing process. The former reflects directly on the medical efficiency of the treatment, while the latter implies a substantial reduction in global costs. Moreover, 3D printing is ideal for the creation of even complex hole patterns in the less-stressed areas of the device, thus performing a topological optimization to reduce weight and enhance skin breathability.



Fig. 5. Patient wearing the 3D printed AFO.

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## AUTOMATIC GUIDANCE OF BIOPSY CATHETER TO PERIPHERAL AIRWAYS TARGETS USING A NOVEL ROBOTIC SYSTEM

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### 1. Introduction

In the process of lung cancer screening, nodules are increasingly detected using computed tomography. Further, although most nodules are benign, tissue diagnostics is required. Several procedures are available nowadays to obtain tissue from peripheral lung nodules: different types of bronchoscopy, confocal laser endomicroscopy or radial probe endobronchial ultrasound. An extensive description of these methods can be found in [1]. All procedures that use image-guided tools expose both the patient and the medical doctor to potentially damaging X-ray.

This paper describes the development of a novel robotic and electromagnetic tracking navigation system, used to guide a catheter in the peripheral airways. Tests of the system, performed on a lung airways phantom, showed that it offers a higher precision and a significant decrease of the X-ray exposure.

### 2. Description of the robotic system

The system has three main components: i. Dedicated software for planning (used to identify the target on the CT of the patient) and navigation (used to navigate inside the airways); ii. Biopsy catheter with pre-bended trackable sensor, and iii. Robotic system (patent pending) with two-degrees-of-freedom (DOF) that inserts and guides the catheter automatically to reach a specific target in the airways (Fig. 1). The robot, controlled by two electric motors, can generate both linear motions (to insert and extract the catheter) and rotational motions (to twist the catheter) and can be used with any type of available catheters and guidewires.

The robot can reach a target without the need of X-ray scanning during catheter insertion, using the AURORA (Northern Digital Inc., Waterloo, Canada) electromagnetic tracking system and a navigation software (iMTECH), developed in our lab. An electromagnetic tracking system sensor with a pre-bended tip is inserted in the working channel of a bronchoscopy catheter, thus tracking its tip during the procedure. When the peripheral target is reached, the sensor is retracted from the working channel to make space for the instrument necessary for biopsy collection (biopsy needle).

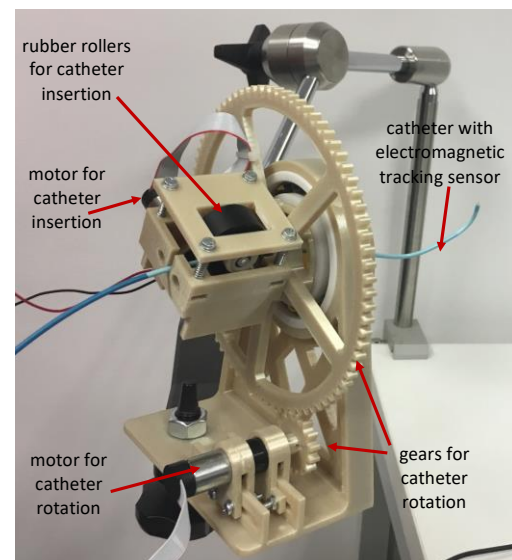


Fig. 1. The robot

### 3. Phantom testing

The system was tested on a silicone phantom of the lung airways, inserted in a plastic box equipped with two electrical motors used to simulate the breathing motion (Fig. 2).

First, a CT scan of the phantom was performed to get the 3D airways model which was subsequently loaded in the software application. The procedure starts by launching the specific software application on the computer and loading the phantom CT data. A fixed reference sensor, attached to the phantom and a mobile sensor, attached to the catheter were used for registration (alignment between the phantom and CT data).

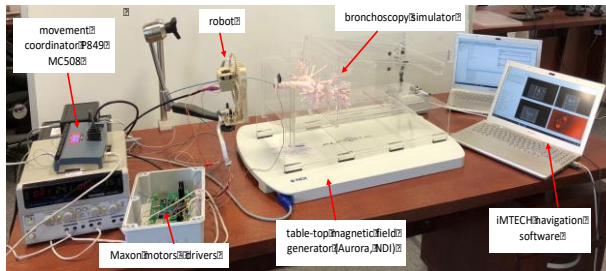


Fig. 1. Set-up for testing of the robotic system

The real bronchoscopy is further planned by the surgeon using the planning module. First, the entry point for the bronchoscope (on the trachea) and the target in the peripheral airways are identified. Then, the software generates a virtual 3D model of the phantom, in which the optimum path between the two points is shown as a continuous line. The procedure for automatic path planning was described previously [3].

After the planning procedure, the doctor commands the start of the robotic navigation procedure in which the robot inserts and guides automatically the catheter in the phantom airways. The tip of the catheter is continuously visualized on the virtual model. The surgeon can correct manually the procedure and can also control the speed. When the catheter reaches the target, the sensor is removed, and the biopsy needle is inserted.

## 4. Results

To test the system, six target points were considered at the end of the small airways on the phantom. Both specialists (medical doctors) and people without medical training (engineers) performed the tests, with and without breathing movement as follows: i. with robot in manipulator mode (using a joystick and checking the position of the tip on the virtual model) and ii. with robot in automatic mode.

In all tests, the targets were successfully reached. Time to target and positioning error with respect to the target are listed in Table 1.

Table 1. Results

|      | Time to target [s] |             |                |             | Position error [mm] |
|------|--------------------|-------------|----------------|-------------|---------------------|
|      | Manipulator mode   |             | Automatic mode |             |                     |
|      | Without breath     | With breath | Without breath | With breath |                     |
| Eng. | 4.5±2              | 11±3        | 13.5±3.5       | 18.5±4      | 5.8 ... 10.9        |
| Dr.  | 3.5±0.2            | 8.1±0.5     | 3.1±0.5        | 15.5±1      | 2.1 ... 5.7         |

One can notice that the procedure in automatic robotic mode was slower. Nevertheless, it has the advantage that it is done at a more controlled and constant speed than in the case of the robot used in manipulator mode.

## 5. Conclusions

The preliminary tests showed that the novel robotic system allows navigation through the bronchial tree, in both manipulator and robotic modes without fluoroscopy scanning. Although the controlled movement of the robotic insertion produces a slower procedure, it is more efficient in the sense that the speed is better controlled, decreasing thus the possibility of injuring the patient due to accidental impact with the airways walls.

A major advantage of the proposed robotic system is the possibility to use any type of catheters available on the market in which an electromagnetic guide wire is inserted, and, after the target is reached, is retracted to give space to the biopsy instrument.

Overall, the new system will allow an easier and more precise guidance of the biopsy devices for targets located in the peripheral airways and will yield cost savings for the equipment in hospitals.

## Acknowledgements

The research leading to these results has received funding from Competitiveness Operational Program 2014-2020 under the project P\_37\_357 “Improving the research and development capacity for imaging and advanced technology for minimal invasive medical procedures (iMTECH)” grant, Contract No. 65/08.09.2016, SMIS-Code: 103633.

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## **SECTION IV: PRACTICAL APPLICATIONS/CASE STUDIES/ INSTRUMENTATION**



# OPTIMIZATION OF LASER BEAM CUTTING PARAMETERS OF HIGH DENSITY COMPOSITE FIBRE CEMENT FLAT BOARD

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## 1. Introduction

Fibre reinforced cement is a composite construction material, which has a growing market in the building industry, because of its strength and durability.

The cutting of a fibre cement board to size by conventional abrasive techniques has several problems. In the case of water jet cutting or chilled sawing, there is a risk of re-wetting of the board, while dry sawing has a very high annual cost of tool wear.

The purpose of this research is to prove the board can be cut with laser beam technology and the other hand to determinate of optimal cutting parameters by Design of Experiments (DoE) method.

## 2. Material and technology

### 2.1 Board material

The fibre cement resists rust, pests, insects and atmospheric effects much more than wooden cladding. It does not warp, its size stability is excellent. Environmentally friendly product. It does not contain or emit harmful gases. The surface does not need to be repainted due to maintenance, therefore its maintenance cost is particularly low compared to other paneling.

The exact composition is protected so that the type of board and its form of the sheet are given [1].

*CEMBIT C 020, Olympus (covered):* CEMENT (65–80%) mostly a mixture of burnt limestone and clay that sets and hardens when exposed to water. PVA (2%) engineered polyvinyl alcohol fibres for reinforcement and to increase product strength, durability and flexibility. FILLER (10–25%) passive, non-binding materials (limestone, microsilica and recycled fibre cement, etc.) to make the mixture easier to process or to add specific properties. CELLULOSE (3–5%) wood fibres help strengthen the product as well as keeping it flexible.

### 2.2 Technological environment

The research was conducted at the Laser Lab and at the Metrology Lab of Edutus College. Manufacturing: TruLaser Cell 7020 5-axis 3D CNC machining cell, TruDisk 4001 Yb:YAG disk laser source configuration, with dual core optical cable.

**Table 1.** Technical characteristics of the CNC cell

| Parameter                               | Value                                 |
|-----------------------------------------|---------------------------------------|
| Wavelength                              | 1030 nm                               |
| Laser power (continuously adjustable)   | 80 - 4000 W                           |
| Optical cable inner/outer core diameter | 100/400 $\mu$ m                       |
| Beam quality                            | 4 mm*mrad                             |
| Power stability (at 4000 W)             | $\pm 1$ %                             |
| Axis travel range X/Y/Z                 | 2000/1500/750 mm                      |
| B/C                                     | $\pm 135^\circ$ / $n \cdot 360^\circ$ |
| Max. axis speed X/Y/Z                   | 100 m/min                             |
| B/C                                     | 90 RPM                                |

Measuring: Mitutoyo Crysta Apex S7106 3D CNC coordinate measuring machine, accuracy:  $1.9+3 \cdot (L/1000) \mu$ m, where L is the length in mm.

## 3. Background and pre-experiment

Our literature work showed that the documentation of this field is virtually zero. We could find one work that has some kind of relationship with our material and with the parameters of the devices in our Laser Lab, but it was not a fibre cement plate cutting, but a laser beam material removal from a concrete block [2].



**Fig. 1.** First cutting tests on 6mm board.

At 2000 W power and 2 m/min cutting velocity with 2 bar N<sub>2</sub> cutting gas the cutting experiment was definitely successful.

## 4. Design of Experiments

Based on the pre-experiment, the following factors and factor levels were determined for DoE:

Optimal (custom) Design  
A flexible design structure to accommodate custom models and minimize the number of runs. Runs are determined by a selection criterion chosen during the build. Factors are treated as categorical.

Categorical factors: 6 (2 to 30) ☐ Horizontal ☒ Vertical

|                 | Name             | Units          | Type    | Levels | L[1] | L[2]  | L[3] | L[4] |
|-----------------|------------------|----------------|---------|--------|------|-------|------|------|
| A [Categorical] | Power            | W              | Ordinal | 4      | 1000 | 2000  | 3000 | 4000 |
| B [Categorical] | Cutting velocity | mm/min         | Ordinal | 3      | 500  | 2000  | 3500 |      |
| C [Categorical] | Gas pressure     | bar            | Ordinal | 3      | 2    | 5     | 8    |      |
| D [Categorical] | Focus distance   | % of thickness | Ordinal | 3      | -90  | -50   | -10  |      |
| E [Categorical] | Frequency        | Hz             | Ordinal | 2      | 6000 | 60000 |      |      |
| F [Categorical] | Gas type         |                | Nominal | 2      | N2   | Air   |      |      |

Fig. 2. Effect factors and factor levels.

The Table 2. shows the output responses:

Table 2. Responses of DoE

| Response                     | Value or Unit |
|------------------------------|---------------|
| Transected                   | yes/no        |
| Burr type                    | pearl/bubble  |
| Linearity                    | mm            |
| Perpendicularity             | mm            |
| Parallelism                  | mm            |
| Deviation from average width | µm            |

The experimental plan was carried out with the fractional factorial, 2-Factor-Interaction (2FI) methodology. The needed and sufficient number of Runs is 66. Specimens are 20\*100\*6mm blocks.

## 5. Results

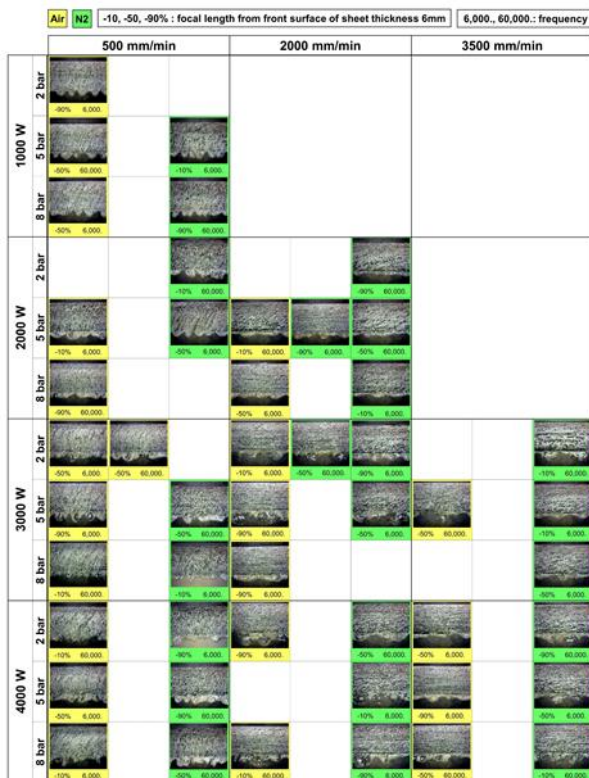


Fig. 3. Success cuts (48 Runs from 66).

For all responses, significant factors can be clearly defined. In all cases the significance level (F-probe, p-value) was less than 0.0002.

Table 3. Optimization conditions

| Name                 | Goal       | Low      | High     | Importance |
|----------------------|------------|----------|----------|------------|
| A:Power              | in range   | 1000     | 4000     | 3          |
| B:Cutting velocity   | in range   | 500      | 3500     | 3          |
| C:Gas pressure       | in range   | 2        | 8        | 3          |
| D:Focus distance     | in range   | -90      | -10      | 3          |
| E:Freq.              | equal to   | 6000     | 60000    | 3          |
| F:Gas type           | equal to   | N2       | Air      | 3          |
| Linearity            | minimize   | 0.0688   | 0.239833 | 1          |
| Perpendicularity     | minimize   | 0.0176   | 5.2958   | 3          |
| Parallelism          | minimize   | 0.1168   | 9.158    | 3          |
| Deviation from Width | target = 0 | -232.896 | 293.304  | 5          |

After optimizing, the Durability=0.871 at A=2kW, B=3500mm/min, C=5bar, D=-10%, E=6000Hz, F=N<sub>2</sub>.

## 6. Conclusions

The fibre cement composite boards can be machined by laser beam technology. The output responses can be controlled by cutting parameters, properly.

As the result of the research, we can give a reasonable estimate to the manufacturers whether it is worth the change their technology, how much it can cost savings or increase production efficiency.

## Acknowledgements

This research was supported by the EU-funded Hungarian grant EFOP-3.6.1-16-2016-00009.

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# DYNAMIC TEST OF A BRIDGE OVER THE DANUBE-BLACK SEA CHANNEL AT AGIGEA

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## 1. Introduction

In order to obtain the dynamic response behavior of a bridge structure some modal analyzes are conducted in the design stage, but the results have to be compared with the experimental response on the real structure. There are some experimental methods available for the in situ dynamic testing of the bridges. In this paper there are presented the experimental tests conducted accordingly with the Romanian standard STAS 12504-86 [1], using test vehicles which induce dynamic excitations in the structure, as in other cases of dynamic bridge testing [2].

The experimental test presume that vertical accelerations of the deck are recorded in real time. The recorded accelerograms are processed in order to obtain the frequencies content of the oscillations through power spectra graphs. In order to validate the experimental results, the frequency content is compared with the numerical results from modal analysis of the bridge structure model.

Taking into account that the bridge is provided with walking decks, beside the structural response testing, the recorded accelerograms were processed through frequency weighting method in order to establish the human comfort responses. The results are compared with the comfort curves from ISO 2631-2 [3].

## 2. Bridge description

The Agigea bridge, km 0 + 540, provides the connection between the North and South area of the port, separated by the Danube-Black Sea Channel. It consists from two viaducts and the main bridge, which provides the over-passage of the channel. This is a cable-stayed bridge with a total length of the superstructure of 362 m. The superstructure has three spans, 80 m, 200 m and 80 m, respectively.

The deck has a hybrid structure, consisting in part of prestressed concrete in the lateral spans and steel beams with RC slab on the rest of the bridge superstructure.

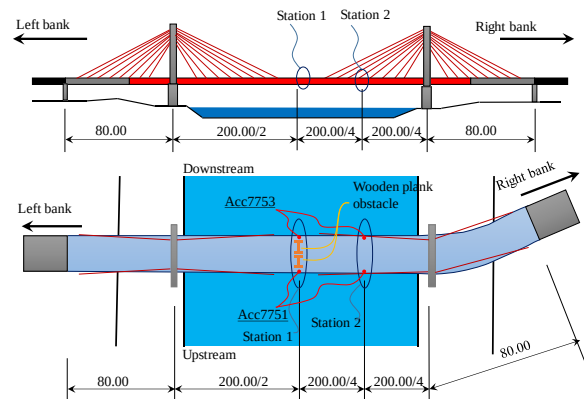


Fig. 1. The cable-stayed bridge diagram and the location of the measuring stations and accelerometers.

## 3. Materials and methods

The vertical accelerations of the bridge deck were recorded in two positions along the principal span of the bridge, on both sides of the bridge. The position of the measuring stations and of the accelerometers is depicted in figure 1.

For the dynamic excitation of the bridge in the experimental tests two loaded trucks were used. The vehicles have been running at a constant speed on the bridge, passing a wooden obstacle positioned in the middle of the main opening. Five speed stages were used for this test, 10 km/h, 20 km/h, 30 km/h, 40 km/h and 50 km/h.

The main principles of the recorded data processing from the dynamic properties point of view and the human comfort response are presented in the technical literature [4][5].

## 4. Experimental result

### 4.1 The identification of the dynamic characteristics of the bridge deck

In order to depict the response of the bridge at the experimental excitation, in figure 2 is presented the recorded acceleration in the upstream accelerometer from station 2 and in figure 3 is the corresponding power spectra of the accelerogram.

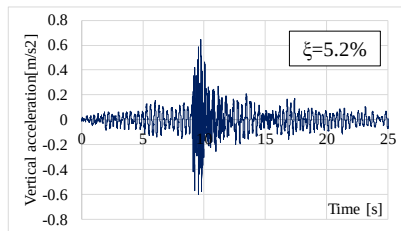


Fig. 2. Vertical acceleration record at 40 km/h.

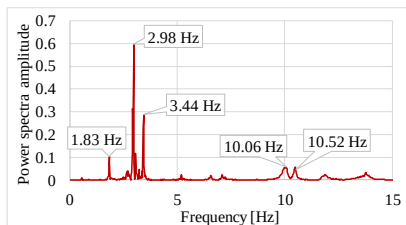


Fig. 3. Frequency content of the accelerogram.

The modal shape with greatest vertical translation participation factor obtained from FEM analysis is 18<sup>th</sup> mode, with a frequency of 2.94 Hz and a 32.6% participation factor.

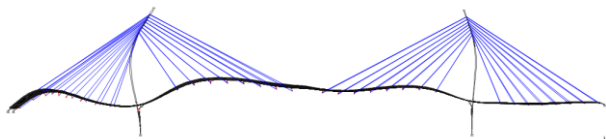


Fig. 4. Mode 18 (2.94 Hz)

### 4.2 The dynamic behavior of the bridge from human comfort point of view

After the frequency decomposition of the accelerogram accordingly with the 1/3 octave spectra, the recorded accelerogram were processed through the frequency weighting method. For a human standing on the walkway of the bridge, one can observe in figure 5 that the behavior of the bridge at traffic-like excitations is almost acceptable, with some slight exceeding of the threshold for 30 and 50 km/h.

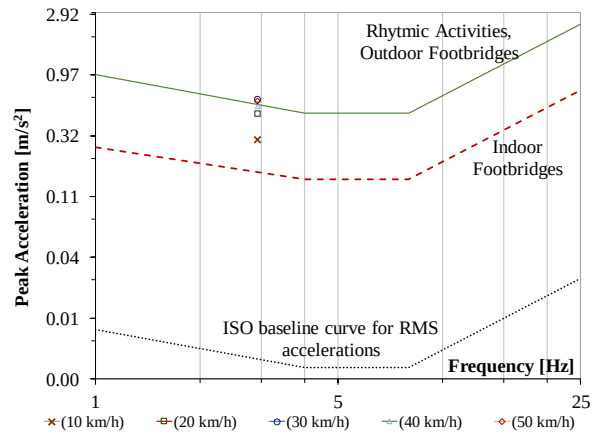


Fig. 5. The peak acceleration comparison with ISO curves.

## 5. Conclusions

The dynamic characteristics of the real bridge are near the values determined from numerical analysis. The most part of the damping properties of the deck is related to the RC slab as one can observe from the average experimental damping ratio of 4.81%. The maximum recorded acceleration was  $1.73 \text{ m/s}^2$  which it is in the threshold accepted for bridges, between 0.30g and 0.4g (around  $3.50 \text{ m/s}^2$ ).

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## SUPERSONIC TEST CASES AT HIGH ANGLES OF ATTACK

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### 1. Introduction

A number of concepts of reusable launch vehicles with flyback capabilities has recently been developed. Also, some modern supersonic air vehicles manoeuvre at high angles of attack. Developments of such projects are supported by CFD (computational fluid dynamics) tools dealing with high angles of attack (high-AoA) high-speed aerodynamics, [1][2].

High-AoA aerodynamics presents many challenges in both experimental and numerical manner. Besides, execution of high-AoA wind tunnel tests may be complicated by constraints of the wind tunnel structures, the dimensions of the test sections, the placements of the model support mechanisms. Therefore, experimental verifications of both the CFD tools and the test setups would be welcome. Standard wind tunnel models are found to be useful and well suited as test cases for such tasks, but the visible sets of reference test results for standard models at high-AoA in the supersonic speed range are relatively small.

Military Technical Institute (VTI) in Belgrade decided to create its own standard-model reference database for future supersonic high-AoA testing and short series of tests of the supersonic-hypersonic HB-2 models were performed at Mach numbers 1.5 to 4 at AoA up to +30°, [3]. A new bent sting, set to 10° bend angle was used as the model support, shifting the AoA range of the T-38 wind tunnel model support from the standard -12°/+20° to -2°/+30°. Obtained data were correlated with results of previous tests performed in the same wind tunnel in the lower AoA range, and with test results from Von-Karman wind tunnels facility of AEDC and limited free-flight data from NASA Ames Pressurized Ballistic Range.

### 2. Standard model HB-2

Hypervelocity Ballistic Correlation Model (HB) in two HB-1 and HB-2 configurations are suitable for supersonic and hypersonic tests. The models were proposed at the joint AGARD/STA meeting in 1959 and adopted in 1960. HB models are based on a blunt cone-cylinder shape with smooth radius fairings, and a 10° tail flare added to HB-2 configuration, Fig.1. All dimensions are given in terms of the forebody diameter  $D$ , [3].

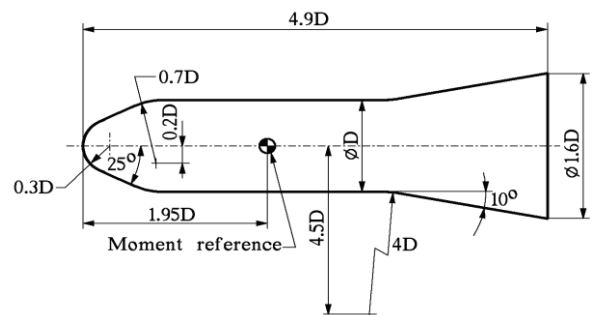


Fig. 1. Definition of the HB-2 model configurations.

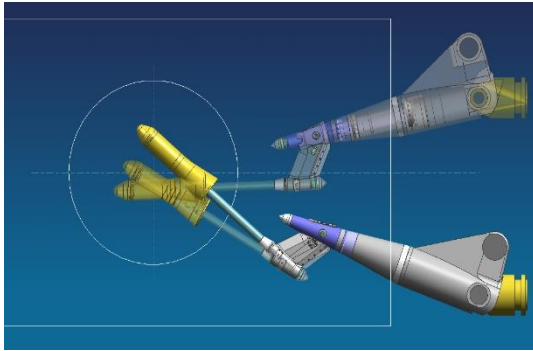
Two HB-2 models were built for tests in the T-38 wind tunnel, with forebody diameters of 75 mm and 100 mm. Using adaptors, both models can be installed on several wind tunnel balances of different load ranges and quickly assembled.

### 3. The T-38 high-AoA model support

Standard model support in the T-38 wind tunnel permits AoA range from -12° to +20° only. A new bent sting was used to shift achievable AoA range.

Bend angle of the sting can be 10° or 20°, set by a key. Achievable AoA range is -2° to +30° on one setting or +8° to +40° on another setting, Fig.2. The straight front part of the sting is exchangeable (alternative parts having interfaces for different force balances) and could be manually set at

different roll angles at 22.5° increments through a full 360° turn.



**Fig. 2.** CAD rendering of the 100 mm dia. HB-2 model on the bent sting used in the tests at extreme AoA.



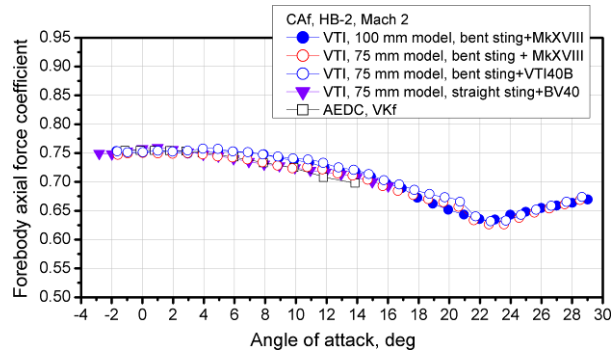
**Fig. 3.** HB-2 model with 100 mm diameter on the 10° bent sting in the T-38 wind tunnel.

## 4. Results

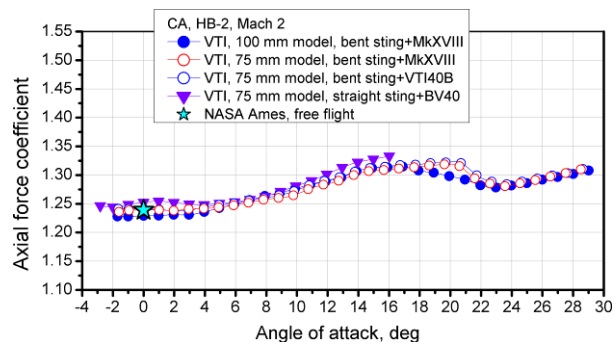
High-AoA tests of the HB-2 model were performed in the Mach number range from 1.5 to 4 and at AoA from -2° to +30° with both models (75 mm and 100 mm forebody dia.), Fig.3. The AoA interval was covered in a continuous movement of the model during the measurement (sweep) at 2°/s rate. The smaller model was tested on a lower-load-range six-component balance designated as VTI40B at Mach numbers 1.5 to 2.5 and on a higher-load-range six-component balance designated as MkXVIII in the complete Mach 1.5 to 4 range. The larger model was tested on the higher-load-range balance only.

In an earlier test in the T-38 wind tunnel the 75 mm model was tested on a straight sting and a special high-drag-range balance designated as BV40 at AoA up to +15°. High AoA data were correlated with these results. Graphs in Fig.4 and Fig.5 show correlated forebody and total axial force coefficients at Mach 2. Available test results from the AEDC wind tunnels and free-flight data from

NASA Ames Pressurized Ballistic Range are presented as well.



**Fig. 4.** Forebody axial force coefficients, HB-2, Mach 2.



**Fig. 5.** Total axial force coefficients, HB-2, Mach 2.

## 5. Conclusions

Tests of two HB-2 models of different sizes, performed in the T-38 wind tunnel of VTI in the Mach number range 1.5 to 4 at angles of attack up to +30°, present the response to a need for standard model data at high angles of attack in the supersonic speed range. Comparison of obtained results with available references for the lower AoA range showed a very good agreement, verifying the new data.

In the lack of available and visible supersonic high-AoA test results for standard models, which are necessary for numerical tools verification, the obtained data are found to be a new contribution.

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# CONCEPT OF A GAS-SENSITIVE NANO AERIAL ROBOT SWARM FOR INDOOR AIR QUALITY MONITORING

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## 1. Introduction

In industrial environments, airborne by-products such as dust and (toxic) gases, constitute a major risk for the worker's health [1]. Major changes in automated processes in the industry lead to an increasing demand for solutions in air quality management. Thus, occupational health experts are highly interested in precise dust and gas distribution models for working environments [2]. For practical and economic reasons, high-quality, costly measurements are often available for short time-intervals only [3]. Therefore, current monitoring procedures are carried out sparsely, both in time and space, i.e., measurement data are collected in single day campaigns at selected locations only [1]. Real-time knowledge of contaminant distributions inside the working environment would also provide means for better and more economic control of air impurities. For example, the possibility to regulate the workspace's ventilation exhaust locations can reduce the concentration of airborne contaminants by 50% [4].

To improve the occupational health and safety of (industrial) workplaces, this work aims for developing a swarm of gas-sensitive aerial nano robots for monitoring indoor air quality and for localizing potential emission sources.

## 2. Robotic Swarm

### 2.1 Robotic Platform

The Crazyflie 2.0 (Bitcraze AB, Sweden) is a flying open-source development platform based on a quadcopter (Fig. 1) [5]. With a take-off weight of 27 g, the Crazyflie 2.0 achieves flight times of up to 7 minutes. The four 4 x 7 mm<sup>2</sup> DC-motors allow a maximum takeoff weight of 42 g, i.e., approx. 15 g remains for additional payload. The diameter of the platform is less than 0.1 m. The Crazyflie 2.0 has a 10 DOF IMU with a 3-axis accelerometer, a 3-axis gyroscope, a 3-axis magnetometer (compass), and a precision pressure sensor for position and altitude control. An expansion

connector (2 x 10 pins) gives access to important interfaces such as I<sup>2</sup>C, UART, and SPI, as well as PWM, analog in/out, and GPIO. A 2.4 GHz data and control link is used for sending control commands and receiving telemetry and payload data. Via Robot Operating System (ROS) or Python SDK, multiple Crazyflie 2.0 can be controlled simultaneously.



**Fig. 1.** The Crazyflie 2.0 nano quadcopter platform (figure taken from [5]).

For the Crazyflie 2.0 numerous expansion boards are available, e.g., an Qi wireless charger (~5 g), a laser-based distance sensor (~1.6 g) for precise height control of the platform, and an optical motion detection system based on an optical flow sensor.

### 2.2 Payload

Aim of this work is the development of a payload unit for gas and airflow sensing in form of a lightweight expansion board for the Crazyflie 2.0 platform. As gas sensor, we consider metal oxide (MOX) gas sensors for monitoring indoor air quality (IAQ), specifically for measuring volatile organic compounds (VOCs). These sensors are characterized by their small size (< 3 x 4 mm<sup>2</sup>) and weight (< 1 g) as well as a response time of few seconds only (e.g., AMS CCS811 or Figaro TGS8100). If detection limits of 1 to 10 ppb are desired, photoionization detectors (PID) could be considered. For example, the PID-AH2 (Alphasense) weighs < 8 g while offering a response time  $t_{90} < 3$  s.

Since airflow information is of high importance for gas-sensitive robots, the airflow vector will be

estimated by fusing the on-board sensors of the Crazyflie 2.0 platform to compute the parameters of the so-called wind triangle [6]. As the applicability of this approach for measuring low speed indoor airflows is unclear (low airflow speeds could be detected as noise only due to compensating movements of the copter), we further consider integrating a hot-wire anemometer from the company Modern Device (Wind Sensor Rev. P, ~2 g) as airflow sensor.

Depending on the scenario, the following payload configurations are considered for the Crazyflie 2.0 platform:

- camera only for visual inspection
- gas sensing only
- airflow sensing only
- combination of gas and airflow sensing

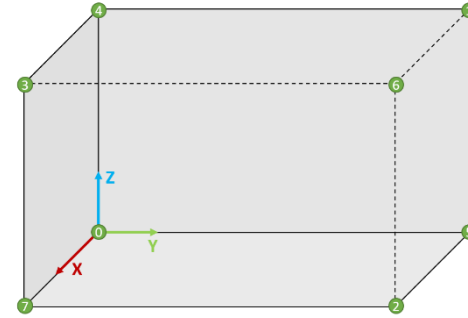
The payload configuration of the different swarm members may vary, i.e., the overall monitoring goal can only be achieved, if cooperative control strategies for swarms are implemented.

### 2.3 Indoor Localization and Positioning System

Bitcraze also offers for the Crazyflie 2.0 platform a local positioning system (Fig. 2), which can be used to estimate the absolute indoor 3D position of the copter similar to GPS [5]. Here, the copter measures its distances to a set of distributed anchors based on continuously transmitted synchronization packets from these anchors. These distances are then used to estimate the absolute position of the Crazyflie 2.0, which can be used for autonomous flight. With the local positioning system, an accuracy of about 0.1 m can be realized. The weight of the corresponding expansion board is only 3.3 g on the copter side, leaving 11.7 g for own sensors.

## 3. Development of Algorithm

Based on this novel indoor air quality monitoring concept, the development and validation of new algorithms in the field of mobile robot olfaction (MRO) are planned, namely gas source localization and gas distribution mapping. For this purpose, multi-agent system (MAS) and particle swarm optimization (PSO) methods will be considered and their suitability for MRO will be evaluated. Furthermore, the development of biologically inspired algorithms and sequential Monte Carlo methods based on the results presented in [7] are planned. Finally, 3D sensor planning strategies and strategies for formation flights are planned to be developed that actively use the rotor downwash of the aerial robots as a large-scale gas transport system.



**Fig. 2.** Reference setup of the indoor positioning system: The green circles are the 8 distributed anchors.

## 4. Conclusion

In this paper we introduce the concept of a swarm of gas-sensitive aerial nano robots for monitoring indoor air quality that aims for improving the occupational health and safety of (industrial) workplaces. Furthermore, we present first starting points for the development of novel swarm based MRO algorithms.

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# PASSIVE RFID TRANSPONDER – LOW-POWER MULTI-SENSOR INTERFACE FOR STRUCTURAL HEALTH MONITORING IN CONCRETE

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## 1. Introduction

Structural Health Monitoring (SHM) has become very important in today's rapidly developing time. High buildings, large bridges and complex technical structures need to be monitored continuously and this over long periods.

Visual monitoring cannot evaluate the internal condition of building structures. Thus, material embedded sensors are needed. Cable connection of these sensors pose the disadvantage of weak spots and water intrusion. For concrete embedded sensors the use of batteries is not convenient, because of limited lifetime, difficult charging, and generation of electrical waste.. Hence, monitoring should be implemented preferentially with firmly embedded passive RFID sensor modules. However, since the concrete structure forms an electromagnetically reflective and absorbing barrier, only limited energy can be transmitted to the module. This project requires a highly energy-saving system, which can record different sensor parameters at critical points.

This paper presents the development and validation of a low-power multi transponder for cascable sensor nodes for material embedded SHM.[1] Further objective of this study is to provide a user-friendly interface for improved on-site monitoring tasks. Section 2 describes the concept of the schematics and the possibilities of the energy managements. Section 3 describes the validation experiments, followed by the results in section 4. The paper concludes with the summary in section 5 and outlook in section 6.

## 2. Concept

To monitor the structure, it is necessary to embed the sensors inside the concrete. For this the distance is important between the embedded transponder antenna and the surface of the concrete, as well as the distance between the surface and the reader-antenna.

It should be noted that, the higher the frequency of the energy and data transmission is, the higher the signal is attenuated by the concrete.[2] Therefore HF RFID can be beneficial for low-power application compared to UHF.[3] Figure 1 shows the concept of the system integration in concrete.

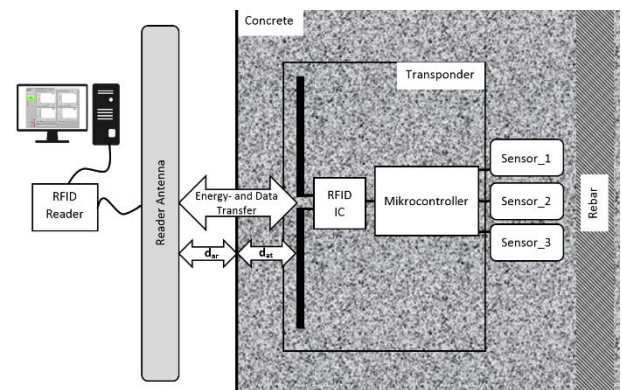


Fig. 1. Concept of the measuring procedure and system integration in concrete.

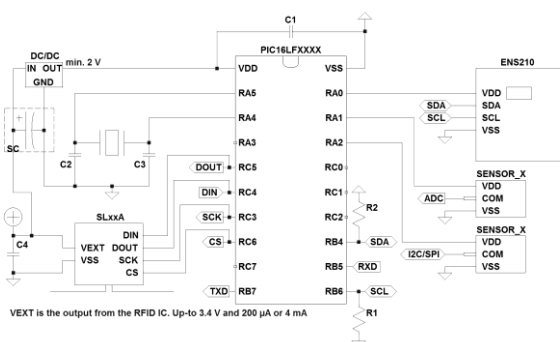
### 2.1 Energy Management

Extremely energy-saving components are required to keep energy consumption low. The Microchip PIC16 (L) F family with eXtrem Low-Power (XLP) microcontrollers ( $\mu C$ ), with a power consumption of  $30 \mu A/MHz$  at  $1.8 V$ , are ideal for this project. The energy requirements of the sensors must be adapted to the system, as only limited energy is available. Depending on technology and frequency, the RFID IC can deliver up to  $200 \mu A$  for UHF and  $4 mA$  for HF, and up to  $3.4 V$ .

The ENS210 (ams) temperature and humidity sensor was selected for the basic transponder. The power consumption of this sensor for one measurement is  $7.1 \mu A/1 MHz$  at  $1.8 V$ . All connected sensors are powered through the microcontroller. It can deliver up to  $50 mA$  at each I/O pin, so that individual sensors can be switched on and off in order to perform defined measurements

## 2.2 Schematics

Figure 2 shows the detailed schematic from the implementation the principle of the sensor connection, and the external wiring of the individual components. Here is important that an external quartz is connected with the  $\mu\text{C}$ . The required frequency is 32.768 Hz. For a better energy-supply, a super cap (SC) is connected behind the supply output of the RFID IC. As mentioned, all sensors are connected to the  $\mu\text{C}$  and receive the energy from the I/O pins.



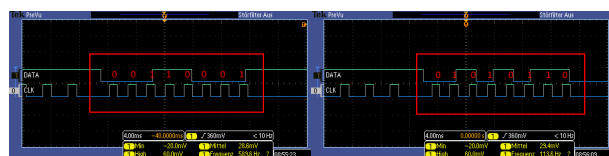
**Fig. 2.** Schematic of the circuit implementation.

### 3. Experimental Setup and Equipment

The actually used laboratory power supply replaces the power supply through the RFID IC. This actual setup ensures a continuous voltage and power supply, furthermore the programming of the  $\mu\text{C}$  is trouble-free. To transfer the source code to the  $\mu\text{C}$ , the IDE provided by Microchip (MPLAB X IDE) can be used with the associated compiler. A programming adapter is also necessary, the PICKit 3 has proven to be highly recommended.

## 4. Results

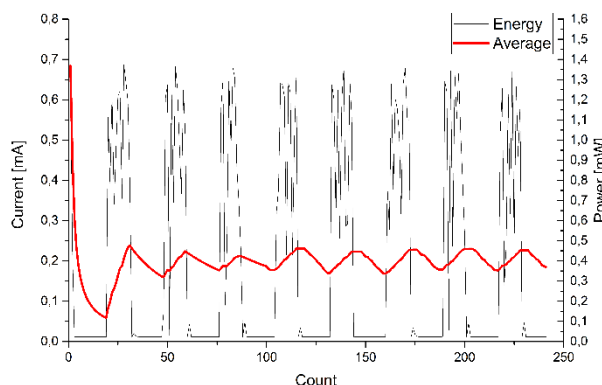
After operating the circuit with max. 2 V voltage, a measurement was performed in the laboratory. The interface to the RFID IC was monitored by a logic analyzer. This interface is used for communication between the  $\mu$ C and the FIFO register of the RFID IC. In this experiment, the value of the humidity sensor was transmitted. The result is shown in Figure 3.



**Fig 3.** Communication monitoring between the RIFD IC and the  $\mu$ C in binary code.

Normal operation can be observed on the left side, with the measured value in binary code of 49 % humidity. In the figure on the right, the sensor was exposed to an artificial moisture source. The changed value in binary code was 86 % humidity.

Figure 4 shows the average current or power consumption of the base system. With the individual measurements and a sampling rate of about 5 seconds, the required power consumption is below 500  $\mu$ W.



**Fig 4.** Average power consumption of the system over 8 measurements

## 5. Conclusion and Outlook

The power consumption between the previous generation and the new generation has been optimized by about 50%. The developed circuit with the corresponding low-power elements shows an enormous potential to become a low-power application in the field of SHM. With the inexpensive construction, commercially available components, and a very low energy requirement several transponders can be installed. Further tests under real environmental conditions are planned and will be implemented soon.

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# COMPARATIVE RESULTS OF MILLING STRATEGIES IMPLEMENTATION

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## 1. Introduction

The paper promote the new cutting methods, in comparison with conventional ones, on CNC machines. The goal was to highlight the advantages of the new cutting methods and the conclusions was obtain after the real manufacturing applications using both strategies.

## 2. State of art

The process of manufacturing planning was presented, following well-defined steps. Was underline the importance of choosing the correct machine tool, cutting tools and the cutting parameters. Four primary technologies determine how efficiently material can be removed during roughing operations:

1. Reasonably rigid CNC machine tools with repeatable feeds and speeds.
2. High-end cutting tools with geometries and coatings created to optimize material removal for specific grades of material.
3. Responsive machine controllers that can adapt quickly to dense sets of CNC program instructions.
4. Material-aware CAM software technology that has the intelligence to generate CNC programs quickly based on actual material conditions ahead of the tool, rapidly adapting the machine and tool motions to maintain a constant chip load according to manufacturers' recommendations.

In case of computer-assisted manufacturing, the strategies used in the manufacturing process also come into calculation. Choosing and correctly applying them can create a more efficient manufacturing process. Taking into consideration the fact that the main goal of the workshops is to

shorten the manufacturing times and to achieve a longer cutting tool life, the optimization of the manufacturing process through different strategies is essential. To achieve these goals, engineers had developed the concept of computer-assisted manufacturing for manufacturing parts on numerically controlled machines.

In order to highlight the importance of the strategies used, in this paper was presented the manufacturing process of a workpiece, by comparing two different manufacturing strategies. For the roughing operations it has been used the dynamic milling strategy, a revolutionary milling method in the field of CNC machining, on one hand, and the conventional pocket milling, on the other hand. To achieve the manufacturing technology, the computer-aided manufacturing program, Mastercam 2018, was used.

The target for applying constant chip load roughing in a workshop is the CAD/CAM software. Dynamic milling brings the required technologies together, making it simple to create roughing programs that use advanced cutting tools.

Tool life is important in any manufacturing applications. Dynamic milling used at constant chip loads according to the manufacturers' recommendations has resulted in up to five times longer cutting tool life.

Dynamic milling uses finesse instead of force, so CNC machines with lower horsepower sometimes can be used. The high torque is not essential when material is removed at less than 50 percent stepovers and deeper stepdowns. This reduces costs by allowing for better usage of existing equipment.

These technologies are now available, and most shops have at least some machines or cells where they can make use of constant chip load roughing strategies. Getting up to speed requires spending a little time learning how they work and how to use

them correctly. Once this low hurdle has been cleared, it is faster and easier to use this approach to roughing.

Based on what your machine can handle, you can progressively decrease stepovers and increase speeds until you arrive at the sweet spot where you have the best balance of material removal volume and tool life. Both will almost always be better than results from conventional aggressive hogging-out strategies that rely on more than 50 percent stepovers and shallow depths of cut.

The software automatically adjusts tool movements that create efficient, a very smooth toolpaths that avoid burying the tools; therefore, the cutting tool, part, and machine are always in a safe cutting condition. It is possible to write cutting programs like this manually, but it takes hours to do it right and then single stepping through the program at the machine to make sure every line of code is correct.

### 3. Conclusions

This advancement in radial chip thinning ensures that the machine tool what are using is operated most efficient. The technology supports a scientific approach to roughing and rest roughing that improves tool life, increases material removal rates, and minimizes programming time. It is easy to implement and safe to use within tooling chip load parameters.

Take in consideration this facts and applied on the real part was obtain:

The time of manufacturing of "open pocket" what was manufacturing with " dynamic milling " strategy was reduced from 199 seconds to the 61 seconds and the "close pocket" made with the same" dynamic milling " in comparison with the conventional pocket milling with " z leavel " strategy was reduce from 110 seconds at 75 seconds.

The total time of manufacturing of the part was reduce with 33% , a substantial time for one manufacturing time.

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# TOW TRUCK FRAME MADE OF HIGH STRENGTH STEEL UNDER CYCLIC LOADING

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## 1. Introduction

Modern materials of high strength are employed for new critical working conditions that are often taking place in many exploited devices and constructions. Among such materials the Strenx Steel plays an important role [1]. The Strenx Steel is recommended for production of special trucks frames. It is manufactured using hot-rolled process. The main advantages of the steel are high elastic limit and ultimate tensile strength usually equal to: 688 MPa and 810 MPa, respectively, Fig. 1.

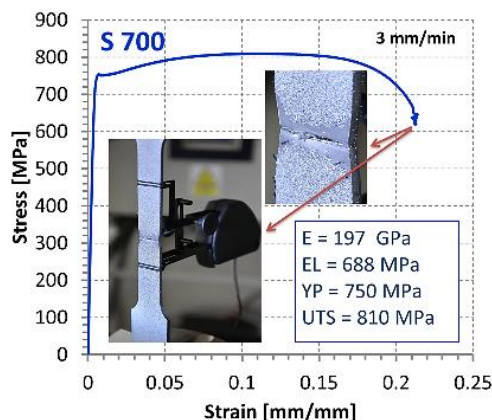


Fig. 1. Tensile curve of the S700 MC steel.

The S700 MC steel can be characterised by the attractive mechanical parameters. A good welding properties are of particular importance. For this type of steel the hardness in the heat affected zone (HAZ) should be lower than that for the parent material, Fig. 2. An importance of HAZ is related to the dominant crack appearance, i.e. the multi-axial stress state components appear in the HAZ, and as a consequence, prevent further crack propagation. Therefore, failure does not take place in the HAZ, but in the parent material or weld [2].

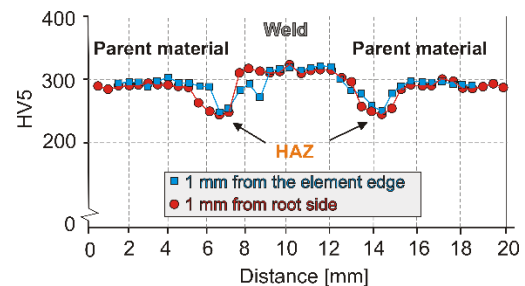


Fig. 2. Vickers hardness distribution for the S700 MC steel after welding, plate thickness 6 mm [2].

The aim of the paper was to examine durability of the special tow truck frame, containing welded joints, (Fig. 3) subjected to cyclic loading.

## 2. Experimental procedure

All tests were carried out on testing stand equipped with servo-motors, digital controller, anti-vibration platform (9 m×3 m) and coupling grip. A force signal in the form of sinusoid was used to control a piston movement. The tested component was placed on the anti-vibration platform. The loading process was executed by means of the spherical grip connecting the piston to the ball of the coupling device A50-X, Fig. 3a, b.

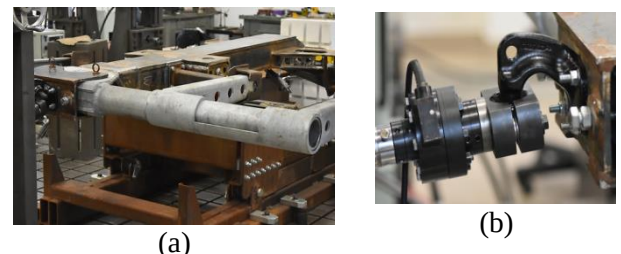


Fig. 3. The 3D anti-vibration platform: (a) general view; (b) connection of the servo-motor and coupling device.

The 3D object (3.5 m×2 m×1.5 m), recommended for pulling of damaged vehicles (trucks, lorries or cars) due to road accidents was tested. It was

mounted to the anti-vibration platform using chassis frame of tow truck by means of screws and clevises, Fig. 3a. Elements of the frame made of the high strength steel (S700 MC) were connected using welding technology.

Parameters of the testing procedure were established on the basis of the 55 Regulation EKG ONZ: amplitude of the force signal -  $\pm 18.6$  kN, frequency - 4Hz, and limit number of cycles -  $2 \times 10^6$ . A slope of the acting force with respect to the horizontal platform was equal to  $15^\circ$ . The frame behaviour was evaluated on the basis of force and displacement variations versus time. Also an optical system was applied for damage development inspections.

### 3. Results

The experiment was carried out up to the frame fracture that has taken place after 573 000 cycles. The crack developed along two paths in the upper and lower parts of the arm. The fatigue fringes were well evidenced at the edge of arm close to the HAZ, Figs. 4, 5.

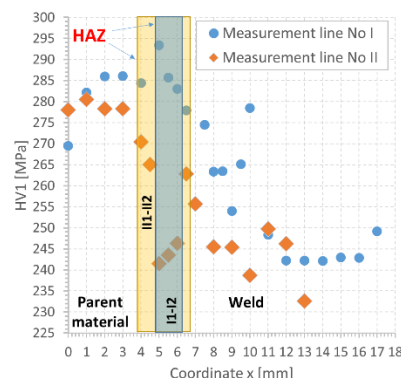


**Fig. 4.** Visualization of the crack in the frame's arm (a) general view; (b) magnified view of the upper part.

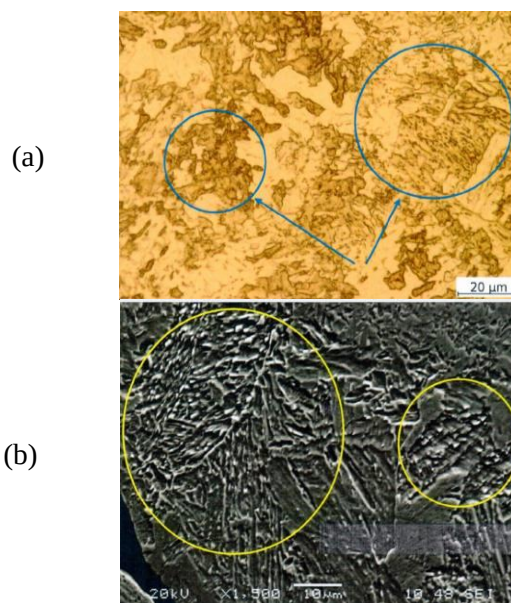


**Fig. 5.** Visualization of the crack in the upper beam: (a) bottom view; (b) fatigue fringes.

On the fracture plane the hardness of the parent material was equal to 285 MPa. It was larger of around 40 MPa than that in the weld section measured. It should be taken into account, that the soft zone was not identified, Fig. 6. Microstructural analysis of the HAZ showed a higher content of sintered carbides in comparison to the parent material. It also revealed their irregular distribution, Fig. 7.



**Fig. 6.** HV distribution in the fracture plane.



**Fig. 7.** The HAZ from microscopical observation: (a) LM 500 $\times$ ; (b) SEM 1500 $\times$

### 4. Summary

Welding of the high strength steel requires more advanced technology than the process for typical steels. The soft zones mainly located in HAZ are important features of the welded region. If the soft zone areas have not an adequate mechanical parameters associated with the microstructure, then the fatigue cracks may develop in HAZ.

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## STATIC TESTS ON A RAILWAY VEHICLE BODY BEFORE BOGIE REPLACEMENT

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### 1. Introduction

A railway vehicle (locomotive, passenger car or freight wagon) can be used by the owner for 20-30 years. During the lifetime of the vehicle, some components may be replaced with redesigned ones, following new technologies and engineering solutions, even if maintenance is done periodically. One element that can undergo such replacement is the bogie.

Firsts bogies for passenger cars were designed as closed frames with axle suspension, car body suspension and spherical center bowl, like the Görlitz bogie (Fig. 1).



Fig. 1. Görlitz bogie.

A more advanced bogie design is the Y32 bogie (Fig. 2). For this one, the chassis of the passenger car needs to be modified in order to connect with the body of the vehicle, because there is no center bowl and the bogie has steering axles; this type of bogie has a better running dynamics behavior.



Fig. 2. Y32 bogie.

The paper presents a static test in which stress analysis was undertaken on a railway passenger car,

before the modifications needed for the replacement of the current Görlitz bogie with a redesigned Y32 one.

All static tests were done in the Rolling Stock Laboratory of the Romanian Railway Authority – AFER, on a test rig. This device can be used to test locomotives, passenger cars, freight wagons with horizontal forces (up to 2000 kN) or vertical loads (up to 60 tones).

### 2. The studied structure

Before the replacement of the bogie, the car body of the vehicle was tested according to the railway standards in order to have a detailed view of stresses for different load cases.

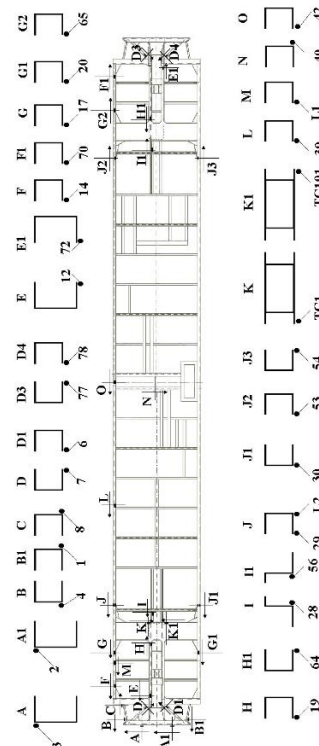


Fig. 3. Detail of the chassis and strain gages.

For this purpose, strain gauges were glued in different areas of the chassis, as shown in Fig. 3, [1].

### 3. Experimental set-up

During tests, HBM Centipede data acquisition systems were used. The software application was specially created and customized for this project in HBM Catman Professional data acquisition software by the AFER authors of this paper. As sensors, HBM 1-LY11 10/120 strain gauges were used.

### 4. Tests

The following static tests were performed [2]:

1. Compressive force of 2000 kN at coupler level;
2. Compressive force 2 x 1000 kN at buffer level;
3. Compressive force of 500 kN applied diagonally at buffer level (right);
4. Compressive force of 500 kN applied diagonally at buffer level (left);
5. Compressive force of 400 kN, at 350 mm above buffers level;
6. Compressive force of 300 kN at the level of the window sill;
7. Compressive force of 300 kN at the level of the cant rail;
8. Tensile force of 1500kN in coupler area;
9. Vertical static loads for the vehicle body;
10. Lifting of the vehicle at the specified lifting position;
11. Superposition of static load cases for the vehicle body.

A picture of the applied loads in test no. 5 is shown in Fig. 3.

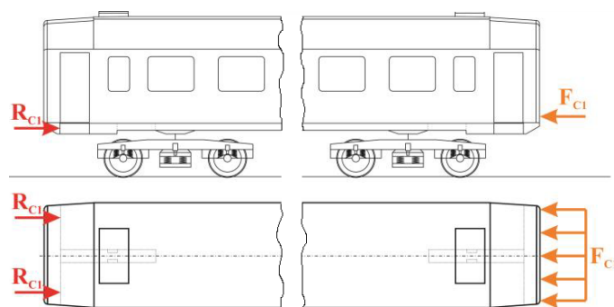


Fig. 4. Loads applied in test no. 5

### 5. Results

Some of the results for the first five tests at different locations of the strain gauges are listed in Table 1.

Table 1. Results of some static tests, stress [MPa]

| Strain gauge | Test |      |      |      |     |
|--------------|------|------|------|------|-----|
|              | 1    | 2    | 3    | 4    | 5   |
| 1            | -84  | 322  | 22   | 139  | 0   |
| 2            | -28  | 37   | 4    | 4    | 1   |
| 3            | -35  | 41   | 5    | 7    | 5   |
| 4            | -64  | 321  | 146  | 18   | 1   |
| 6            | 92   | 268  | 58   | -35  | -3  |
| 7            | 80   | 215  | 27   | -42  | -4  |
| 8            | 0    | -21  | -2   | 0    | -1  |
| 12           | 215  | 309  | 4    | -172 | -10 |
| 14           | 222  | 301  | 8    | 178  | -25 |
| 17           | -70  | 124  | -59  | 12   | -6  |
| 19           | -239 | -135 | -17  | -33  | -11 |
| 42           | -273 | -229 | -47  | -39  | -26 |
| 53           | -254 | -298 | -21  | -79  | -32 |
| 54           | -255 | -272 | -75  | -16  | -39 |
| 56           | -105 | -108 | -28  | -11  | -15 |
| 64           | -212 | -137 | -35  | -23  | -24 |
| 65           | -88  | -126 | -68  | 13   | -7  |
| 70           | -198 | -319 | -141 | 9    | -31 |
| 72           | 132  | -321 | -144 | -4   | -13 |
| 77           | 53   | -190 | -33  | -30  | 9   |
| 78           | 78   | -204 | -27  | -40  | 14  |
| L1           | -190 | 176  | -14  | -49  | -14 |

The allowable stresses for the material of the chassis (steel S355) are presented in Table 2, [2]:

Table 2. Allowable stresses for the considered material

| Stress           | Welding free area | Welding area |
|------------------|-------------------|--------------|
| Horizontal [MPa] | 355               | 329          |
| Vertical [MPa]   | 355               | 329          |

### 6. Conclusions

As one can notice from Table 1, the obtained stresses are below the allowable values for the material of the chassis.

No residual stresses or strains were recorded after unloading in the performed tests.

This was the first step to modify the chassis of the vehicle. Based on the obtained, a finite element model was calibrated and the body of the car was redesigned in order to replace the old bogie with a new one. The redesigned car was also tested by AFER, both statically and dynamically.

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- [2] \*\*\*SR EN 12663-1/2010 Railway applications. Structural requirements of railway vehicle bodies. Locomotives and passenger rolling stock (and alternative method for freight wagons).

## EXAMINATION OF THE INTERMODAL TRAILER FRAME IN ACCORDANCE WITH UIC 596-5

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### 1. Introduction

The aim of work was to perform a comparative analysis of the results of the examination of the intermodal trailer frame structure. The tests were conducted with the Finite Element Method (FEM) [1, 2] in the static range. The analysis included the simulation of the test of trailer lifting by the two handles on either side of the frame, according to the UIC 596-5 standard.

The results obtained from the numerical simulation were compared with the results of the experimental studies. The research involved the determination of the value of displacements in selected points of the load-bearing structure of the intermodal semitrailer. The measurement of the deflection values was made by means of photogrammetric method using the dedicated system TRITOP of GOM company [3].

The tests consisted of performing the deflection measurements, under the load resulting from the even distribution of the load on the surface of the trailer.

### 2. Experimental testing

The position of the load in the form of single loads on the frame is shown in Figure 1.



Fig. 1. Position of the load situated on the trailer.

The requirements concerning the method of testing the strength of the frame structure of the intermodal trailer are defined by UIC 596-5, in Annex A - "Measurements and arrangement of grab handles on semi-trailers" attached to the standard. The test of lifting the entire frame with the load is presented in Figure 2.



Fig. 2. Lifting of the trailer with a load.

The loading of the semitrailer during testing is important because of the effort determination for the particular elements of the trailer. According to the guidelines, the load is derived from the mass forces, which values is determined on the basis of GVM

(gross vehicle mass). In the case of the analysed trailer, the GVM is 41T. The load value was determined in accordance with UIC 596-5 and it is equal:  $1.25 \cdot \text{GVM}$ . During a test, lasting for at least 5 minutes, no damage may occur, consisting of permanent deformations and changes that would prevent its further use as intended.

The total value of the loads from mass forces adopted for analysis is 51.25T.

The results of the experimental analysis are shown in Figure 3.

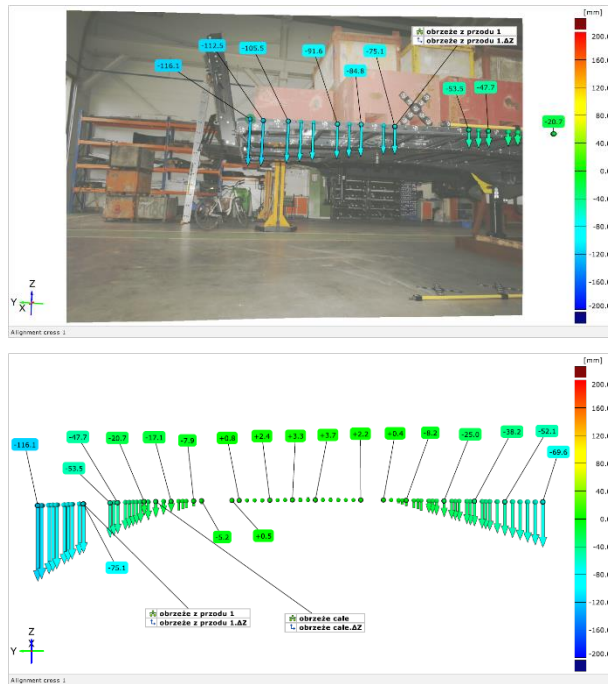


Fig. 3. Results of the experimental examination.

### 3. Numerical calculations

On the basis of the geometric model of the trailer frame, a discrete model was created for the FEM calculations [3]. The structural elements were modelled using eight-node solid elements (tetragonal elements were used only to model parts of the suspension brackets). Welded joints in the trailer have been modelled by kinematic coupling of the corresponding surfaces of the connected elements, thus obtaining the continuity of the displacement field. In the case of fillet welds and butt welds, in which the distance of joined elements was considerable, rigid intermediate elements (beams) were used to join the surfaces [4].

The model of the material used in the analysis is a linear elastic steel, where:

- Young's modulus:  $E = 210 \text{ GPa}$ ,
- Poisson's ratio:  $\nu = 0.3$ .

The results of the numerical analysis are presented in Figure 4.

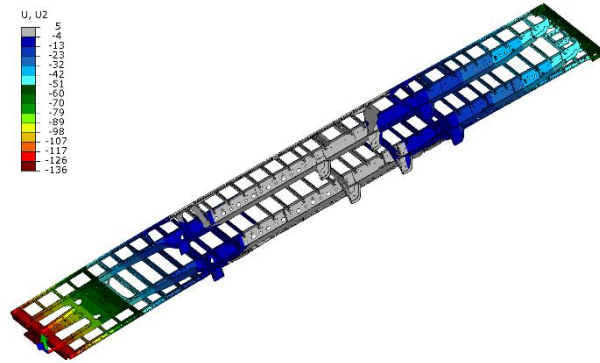


Fig. 4. Results of the numerical analysis.

### 4. Results comparison

The maximum bend deflection during the experimental test occurred in the front part of the trailer frame and amounts to 116mm. In the case of numerical analysis, the maximum deflection occurred in a similar place and was around 126mm.

Data regarding deflections obtained as a result of the numerical simulation are convergent with the data obtained from the photogrammetric measurements. It should be noted that as a result of the simulations, deflections are about 10% higher than those measured in the experiment. This is due to the adopted algorithm of solving the problem. In the finite element analysis, geometric nonlinearity was not taken into consideration.

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## STRENGTH OF BRIDGE STRUCTURES: A STUDY

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### 1. Introduction

At the beginning, these structures were mainly made in wood and stone, with reduced lengths. The development of human capacity and means made it possible to build real monuments anchored in steel pillars and cables.

Therefore, the purpose of our work is to highlight the evolution of bridge structures. We chose pasta as a building material. These have an impressive weight to strength ratio. To get a better structural performance, we designed a solid structure using a simple material that will be tested at a shearing force.

### 2. Materials and methodology

#### 2.1 Materials properties

In elaborating the structure of the bridge we used spaghetti because they have an annular section.

They have the following properties at normal temperature and humidity:

- Maximum tensile strength: 140 MPa;
- Stiffness (Young's modulus):  $E=703069$  MPa;

In comparison, cast aluminium(wet or dry):

- Ultimate tensile strength: 703 MPa;
- Stiffness  $E=10306$  MPa;[1]

This material has a high resistance on the longitudinal direction. A piece of spaghetti with a 2 mm diameter can support up to 4.5 kg. A very important notion is that the maximum load does not depend on the length of the pasta. The maximum load is equal to the tensile strength multiplied by the section.

The spaghetti have a low compressive strength, but if we divide their length by half, they can hold up to 4 times the initial force. Release can be:

- If  $L/d < 10$ , failure is by crushing
- If  $L/d > 10$ , failure is by buckling[1]

The constant of proportionality  $k$  can be determined as follows:

$$F = k \frac{d^4}{L^2} \quad (1)$$

Where  $F$  is the force required to crush the pasta by compression,  $d$  and  $L$  the diameter, respectively the length of the pasta.

In order to determine the constant of proportionality  $k$  the following steps are made[1]:

- Measure length and diameter of a piece of spaghetti
- Hold spaghetti vertically on postal scale
- Press down on spaghetti until it begins to bend
- Read load  $F$  on postal scale
- Calculate  $k$ .

#### 2.2 Structural elements

The lower part of the bridge was made by gluing pasta in a longitudinal section. As a connector, we used a hot glue gun. The glue was applied everyone centimeter, it was not spread across the joint between the pastes to avoid excess leakage. For the experimental part, we made two bridge models[2]. The first one made from five rectangular cells of 80 mm length and 70 mm in height with a diagonal of 106mm (Fig.1) and the second one was made of 5 isosceles triangles with a base of 100mm , 100mm height and two sides of 110mm (Fig.2)[3].



Fig. 1.

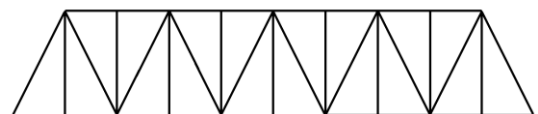


Fig. 2.

The two ends of the bridge are represented by two triangles with a base of 80mm and a height of 70mm and respectively 50mm and 100mm (Fig.3). This shape of the ends was chosen not only for design purposes but also to be able to withstand the weight at which the bridge is subjected to more efficiently.

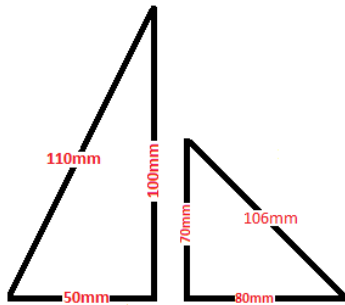


Fig. 3. The dimensions of the bridge ends.

As a means to hold the weight we used a hook that was attached to a piece of PAL wood that has the size of 12x3.5 cm. It was attached using 2 screws and nuts (Fig.4). We used the piece of PAL so that the force would be evenly distributed on the bridge as much as possible.



Fig. 4. The piece of PAL with the hook

For the experimental study we built a structure with a length of 600mm and a height of 100mm, built from triangular cells whose angle with the base was 60. We built a continuous surface for the bridge base in order to distribute the force. For the same reason we added a piece of PAL, that had at the same time the role of a holding mechanism. We placed the bridge in between 2 pillars that allowed us to simulate the load of a real bridge. The weight that we applied to the structure was a bucket which we gradually filled with water until the structure failed.



Fig. 5. Testing of the structure

### 3. Results

The results of this experiment proved, on the contrary, that the structure formed of triangular elements was a more stable and rigid one, weighing only 210 grams and sustaining about 4.5 kg.

### 4. Conclusions

In spite of poor mechanical properties of pasta, we can apply on them a higher weight when they are part of a more complex structure (Fig.6).

If pasta can withstand a much heavier weight than their own, then other materials such as steel, that has better properties with a much higher strength/weight ratio, can sustain considerably higher loads if integrated into elaborate structures[3].



Fig. 6. Structure model

### References

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# 1:60 SCALED EARTHQUAKE TESTING OF THE IMMERSED TUNNEL OF THE HONG KONG-ZHUHAI-MACAO-BRIDGE

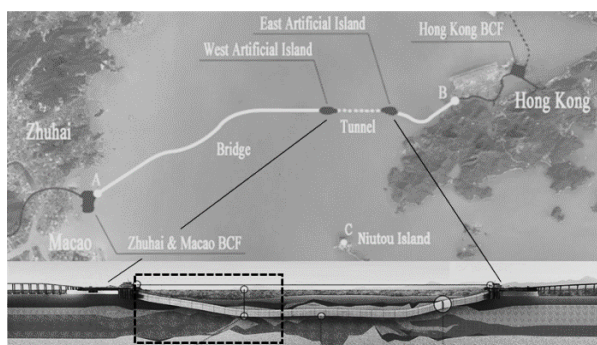
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## 1. Introduction

The Hong-Kong-Zuhai-Macao-Bridge is an approximately 50 km long traffic link between Macao and Zhuhai on the west side and Hong-Kong on the east side of the Pearl River delta (Fig. 1). The bridge is interrupted by a 6 km long immersed tunnel, connecting two artificial islands.



**Fig. 1.** Layout of the traffic link referred to as the Hong Kong-Zhuhai-Macao Bridge (HZMB) and longitudinal section through the immersed tunnel connecting the two artificial islands of the link; the dashed box highlights the part of the tunnel simulated in the earthquake tests.

The immersed tunnel consists of 33 precast, reinforced, and pre-stressed concrete elements. They were immersed, one after the other, into a trench at the bottom of the sea, see [1] for a more detailed description of the construction process. The seismic activity in the area of the immersed tunnel is rather moderate. Still, the expected peak acceleration in Zhuhai and Macao amounts to  $0.98 \text{ m/s}^2$  and in Hong Kong to  $1.47 \text{ m/s}^2$  [2].

The joints between neighboring elements are sealed with two types of rubber rings. The primary sealing ring is referred to as “Gina-element”. The secondary sealing ring represents a redundant safety measure. It is referred to as “Omega-element”. The serviceability of the tunnel requires that the joints remain water tight even in the exceptional load case of an earthquake. In this context, it is noteworthy that relative displacements between neighboring

elements are constrained by shear keys and bolts. As for ensuring the water tightness of the joints, the relative displacements in the axial direction must be smaller than a specified tolerance. This was the motivation to perform scaled earthquake experiments at the worldwide largest shaking table test facility, at Tongji University, in Shanghai, China, see [3,4,5].

## 2. Test setup

A 1:60 scaled model of 13 elements of the immersed tunnel was tested, see the dashed box in Fig.1. The reinforced and pre-stressed concrete elements were represented by scaled aluminum elements. The scaled sealing rings were made of rubber, exhibiting a dynamic behavior similar to that of the real Gina elements. The silty sand of the Pearl River delta was simulated by a carefully designed mix of sand and sawdust. The small-scale model was 40 m long. It was resting on four active shaking tables and on so-called passive tables, bridging the gaps between the active tables. Details regarding the design and setup of the shaking table tests are discussed in [5,6,7]. This includes also details regarding the scaling relations, e.g. the geometric dimensions scaled 1:60, the acceleration 5:1, and the frequency 17.32:1.

### 2.1 Test cases and instrumentation

In 44 different earthquake tests, the scaled model of the immersed tunnel was subjected to different excitation scenarios. As regards *qualitative* properties of the prescribed acceleration histories, three different earthquakes were simulated, see the first column in Table 1. As regards *quantitative* properties of the three prescribed earthquakes, the acceleration histories were amplified such that they refer to three different earthquake intensities, see the second column in Table 1 for the corresponding small-scale acceleration values. The shaking test facility allows for simulating different travelling directions of the earthquake waves and different

polarization directions. The earthquake waves were simulated either to travel from west to east, or from east to west, or a uniform excitation was simulated, see the third column in Table 1. In the latter scenario, all four shaking tables were forced to vibrate identically. The shaking tables can either be moved in the direction perpendicular (“transversal”) or parallel (“longitudinal”) to the tunnel axis, or in both directions at the same time (“biaxial”), see the fourth column in Table 1.

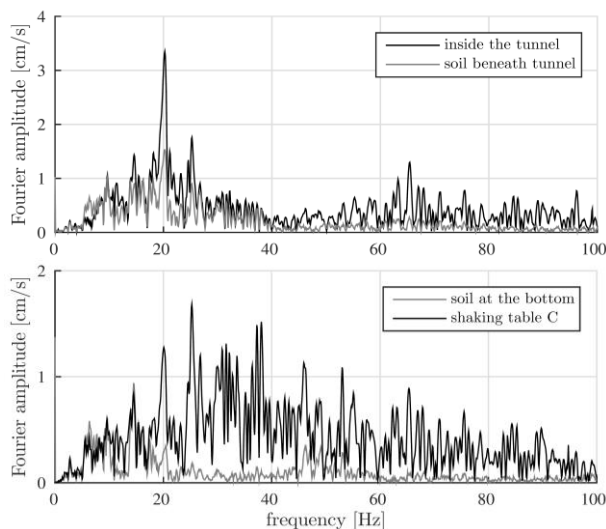
**Table 1.** Different excitation scenarios

| Earth-quake | Max. accel. | Travel direction | Excitation direction |
|-------------|-------------|------------------|----------------------|
| Kobe        | 0.25 g      | west to east     | longitudinal         |
| El Centro   | 0.75 g      | east to west     | transversal          |
| Artificial  | 1.10 g      | uniform          | biaxial              |

The dynamic behavior of the model was recorded by 27 accelerometers and 24 displacement sensors. The relative displacements across all 12 joints were measured with two displacement sensors each. 11 accelerometers measured the longitudinal acceleration of the central tunnel elements. The remaining 16 accelerometers were positioned inside the soil.

### 3. Test evaluation and results

The measured acceleration histories are subjected to a Fourier transform, see Fig. 2.



**Fig. 2.** Fourier transforms of selected acceleration readings recorded during the simulation of the El Centro earthquake, traveling from west to east, with a maximum small-scale acceleration of 0.25 g.

The Fourier amplitude spectra allow for a comparison of the dynamic behavior measured at different positions of the small-scale tunnel. Fig. 2, for instance, shows that small-scale acceleration

frequencies larger than 20 Hz, which is equivalent to real-scale frequencies larger than 1.15 Hz, are damped effectively by the soil. It is also interesting that the soil showed rather small amplitudes referring to small-scale acceleration frequencies around 20 Hz, while the maximum amplitude of the tunnel element was recorded at that frequency. This suggests that the eigenfrequency of the real-scale tunnel is on the order of magnitude of 1 Hz.

The Fourier transforms of the readings of all 27 accelerometers, captured during all 44 different simulation cases, provides valuable insight into the dynamic behavior of the soil and the tunnel during earthquakes, and they allow for a statistical analysis.

### Acknowledgements

Financial support by the Austrian Science Fund (FWF), provided within project P 281 31-N32 "Bridging the Gap by Means of Multiscale Structural Analyses" is gratefully acknowledged.

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# ASSESSMENT OF GRINDING WHEEL WEAR VIA MONITORING OF PROCESS POWER

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## 1. Introduction and motivation

Grinding is a process used to manufacture parts with tightly tolerated geometric features and high demands concerning quality and hardness of functional surfaces. Typically, grinding is one of the last stages of the process chain. Thus, most of the manufacturing costs have already been invested in the part through other process steps like casting, forging, milling or turning and hardening.

Therefore, the supervision of the grinding process is of high interest for manufacturers in order to maintain high part quality over time, e.g. with increasing tool wear. A continuous monitoring of the tool wear state can minimize the risk of tool failure and reject parts as well as maximize dressing intervals and thus productivity [1].

The grinding wheel wear state can be determined directly, e.g. by optical observation, tactile probing or the supervision of magnetic properties [2, 3]. However, production needs to be interrupted for direct wear assessment and certain methods are elaborate or even impossible to implement in situ.

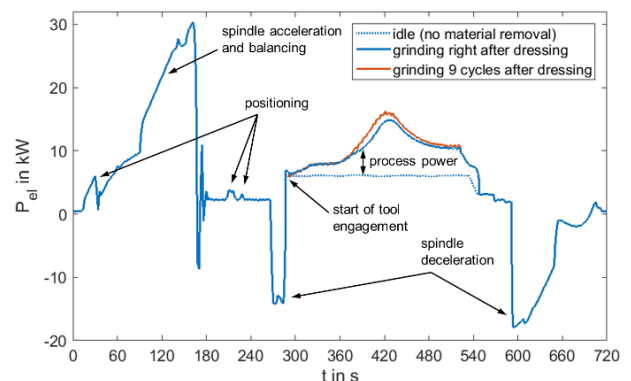
Thus, indirect methods to determine wheel wear are beneficial as they do not influence the cycle time. Following exemplary indicators are suitable for on-line monitoring of the wear state of grinding wheels: cutting force, acoustic emission, vibration, temperature and electrical current or power [2-4]. Dynamometer systems used to monitor cutting forces with a high temporal resolution are costly and laboratory conditions may be required to generate accurate results [4]. To assess the wear state through acoustic emission, vibration or temperature, robust sensors have to be placed as near as possible to the tool engagement zone which typically requires extensive effort concerning design and signal processing [2, 3].

Therefore, this study presents a novel method to monitor wheel wear via a determination of process power. It is based on acquisition and processing of electrical power consumption signals of grinding spindle and feed drives and is comparatively easy to

implement. The used signals are either measured with dedicated sensors or acquired directly from drive controllers. The method builds on an approach by Hacksteiner et al. who showed for different turning and milling processes that process power rises with progressing tool wear and developed an on-line process power monitoring system based on drive controller data acquisition [5].

## 2. Methodology

In general, process forces during grinding increase with increasing wheel wear [6, 7]. Fig. 1 shows the electrical power consumption of a GST CNC grinding machine drive system during one manufacturing cycle of a cylindrical rotor shaft grinding process using a corundum grinding wheel. The power consumption of the drive module (supplying both spindle and feed drives), which was measured using a three phase power meter, rises strongly during tool engagement compared to load-free condition. This difference, representing the process power (i.e. that necessary for material removal), rises with progressing tool wear. However, the measured signal had to be smoothed for improved visualization of the process behaviour and the increase is rather weak which may complicate reliable tool condition monitoring.



**Fig. 1.** Power consumption of the drive system of a CNC grinding machine for a given series manufacturing process (cylindrical grinding of a rotor shafts)

Therefore, a refined approach is pursued to determine tool wear from certain process power and

energy components, respectively. According to Malkin et al., grinding wheel wear shows a stronger dependence to the normal component  $F_{c,n}$  of the cutting force as to the tangential component  $F_{c,t}$  [6]. Former is related to the feed drive process power  $P_{P,ax}$  and the feed velocity  $v_f$  and latter to the spindle process power  $P_{P,sp}$  and the cutting speed  $v_c$ , respectively:

$$F_{c,n} = \frac{P_{P,ax}}{v_f} \quad (1) \quad F_{c,t} = \frac{P_{P,sp}}{v_c} \quad (2)$$

To determine the process power  $P_{P,i}$ , the electrical power consumption of the respective drives in non-loaded condition (but at the same speed) is subtracted from that during grinding:

$$P_{P,i} = \eta P_{el,i} = P_{el,i \text{ load}} - P_{el,i \text{ idle}} \quad (3)$$

This is done in order to eliminate the influence of electrical and mechanical losses in the drivetrain and such due to drag by high-pressure cooling lubricant supply to the grinding wheel (cf. [8]). The actual power consumption during processing has to be determined via direct power consumption measurement or according data acquisition from the drive system. On the other hand, that in non-loaded condition may either simply be recorded and taught-in for a given process or determined from actual speed or feed rate data via mathematical models of the power consumption of according drives (cf. [5]).

Depending on the process type, additional drive signals may be evaluated in order to gain reliable information, e.g. that of the work piece spindle for cylindrical grinding or that of the transverse feed drive for face grinding.

### 3. Results

The monitoring approach is currently tested on cylindrical grinding processes for series production. As an example, Fig. 2 shows the process power evaluated from measured power consumption of the grinding spindle and feed drive of a JUNKER CNC grinding machine during cylindrical grinding of 6-cylinder crankshaft main bearing seats using a CBN wheel. Even though feed drive process power is much lower compared to that of the grinding spindle, both processed signals show good potential concerning the planned assessment of wheel wear and therefore process and work piece integrity. However, the effects of wheel wear on the acquired signals and according implications on work piece quality (surface quality and dimensional accuracy) have yet to be studied in detail.

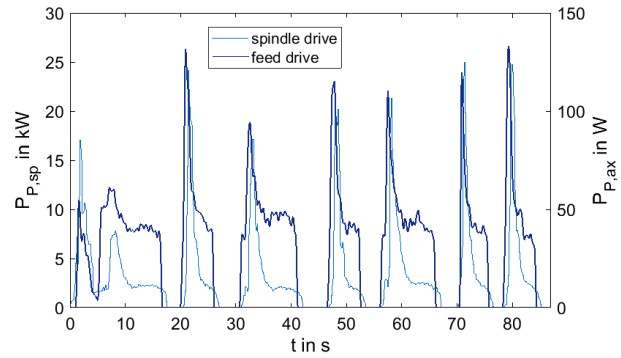


Fig. 2. Spindle and feed drive process power signals during cylindrical grinding of crank shafts

### 4. Conclusions and outlook

This work presents a method to indirectly monitor wear of grinding wheels via spindle and feed drive power consumption. The approach, which is currently tested on series manufacturing processes, is comparatively easy to implement for a given grinding process and machine, respectively. It could facilitate an optimized utilization of grinding wheels and thus help to decrease reject rate and cycle time.

However, further work has to be carried out to identify the best data acquisition and processing strategy for the given industrial applications as well as to prove sufficient sensibility of the method.

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# MULTI-INTERVAL LOAD CELL WITH STEPLESS AUTOMATIC RANGE TRANSITION ACCORDING THE WEIGHT

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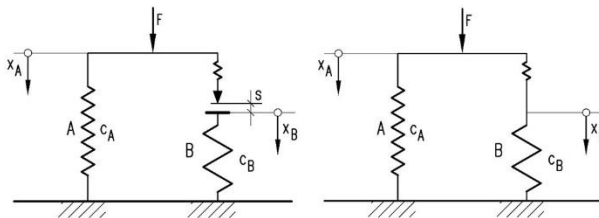
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## 1. Introduction

In the field of weighing technology, a distinction is made in single- and multi-interval systems, [1]. Multi-interval instruments have two or more partial weighting ranges, each with different scale intervals. The paper describes a new designed two-interval load cell system with stepless automatic range transition. The system consists of two coupled load cells with a transition point of 2.5kg. The maximum load is 300kg.

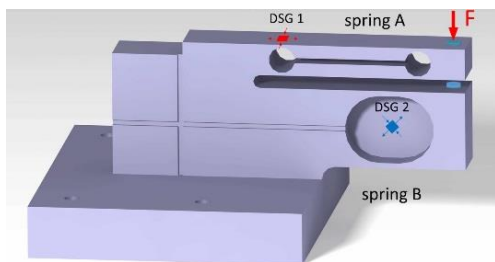
## 2. Operating principle and construction

Figure 1 shows the operating principle in form of a spring model. In the first range, fig. 1 left, only the spring A is in action and the sensor on A gives the output signal. After closing the mechanical contact, fig. 1 right, both springs are in action and the output signal results from sensors on A and B.



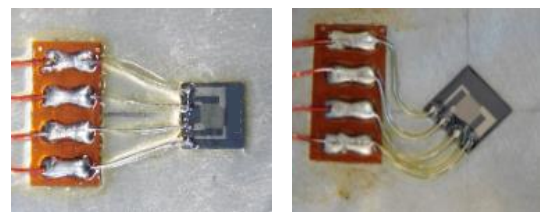
**Fig. 1.** Spring model of the two-interval scale; Left: contact open, spring A in action; Right: contact closed, springs A and B in action

Figure 2 shows design and implementation of this operating principal. Spring A is a double bending beam and spring B a shear beam.



**Fig. 2.** Load cell consist of springs A and B, position and orientation of the differential strain gages

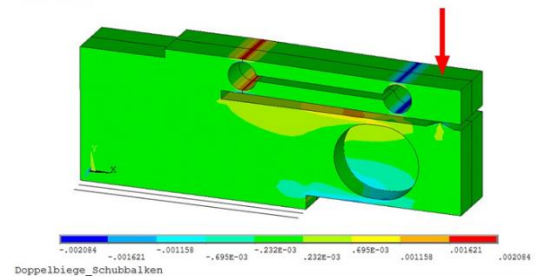
Differential strain gages DSG 1 and DSG 2 [2], figure 3 are installed in the deformation zones of spring A and B. They transform the deformation state in output voltages.



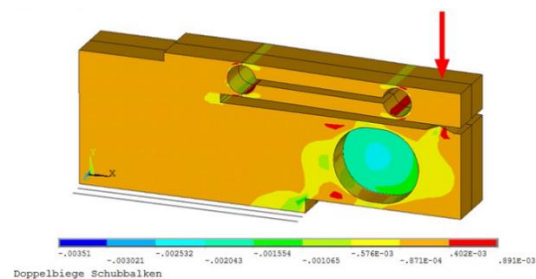
**Fig. 3.** Differential strain gages DSG 1 and DSG 2

## 3. Numerical simulations

At the stage of development and design many numerical simulations using the finite element method ANSYS 17.1 were performed. Selected results are shown in figures 4 and 5.



**Fig. 4.** Numerical simulation, normal strain

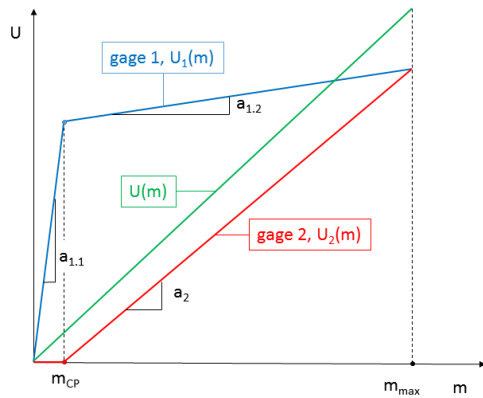


**Fig. 5.** Numerical simulation, shear strain

## 4. Calculation of the mass

The calculation of the mass on the basis of two output signals should be done stepless with automatic range transition in the moment of contact

closing between spring A and B. Figure 6 shows at first both output voltages as functions of the applied mass, blue line  $U_1$ , red line  $U_2$ .



**Fig. 6.** Output voltages  $U_1(m)$  of the gage 1,  $U_2(m)$  of the gage 2, resulting voltage  $U(m)$  calculated according equations (1), (2) and (3)

The calculation of the resulting output voltage, green line is done using equation (1) where the factor  $A$  results from the equations (2) and (3). The parameters  $a_{1.1}$ ,  $a_{1.2}$  and  $a_2$  represent the gradients of the strait lines according to figure 1.

$$U(m) = A \cdot U_1(m) + U_2(m) \quad (1)$$

$$A \cdot a_{1.1} = A \cdot a_{1.2} + a_2 \quad (2)$$

$$A = \frac{a_2}{a_{1.1} - a_{1.2}} \quad (3)$$

For the determination of mass, we introduce the calibration factor  $B$  and can calculate the resulting mass in the first range up to the contact point  $m_{CP}$

$$m_I = B \cdot A \cdot U_1, \quad (4)$$

and in the complete measuring range up to  $m_{max}$

$$m = B \cdot (A \cdot U_1 + U_2). \quad (5)$$

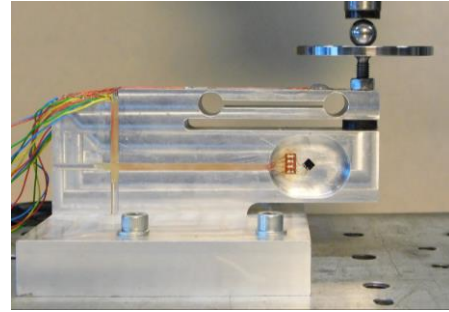
It is recommended to divide the mass calculation in two ranges according equation (6) (Catman Easy Syntax) because the resolution and the accuracy in the first range is much higher. The range transition point (here  $0.9 m_{cp}$ ) should be chosen slightly below the contact point  $m_{CP}$ .

$$mass = if(m < 0.9m_{cp}(m_I, m)) \quad (6)$$

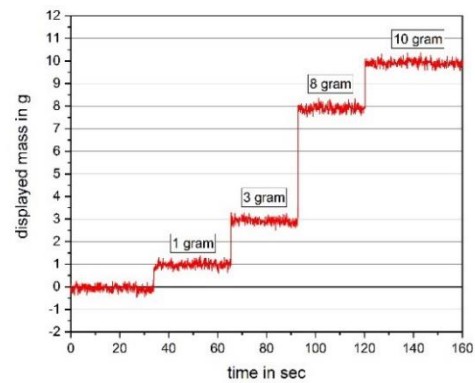
## 5. Test setup and results

The test was performed by applying single masses (1g – 2000g) and by means of continuous loading (0 – 300kg) in a Zwick/Roell testing machine, using a high precision load cell 5kN.

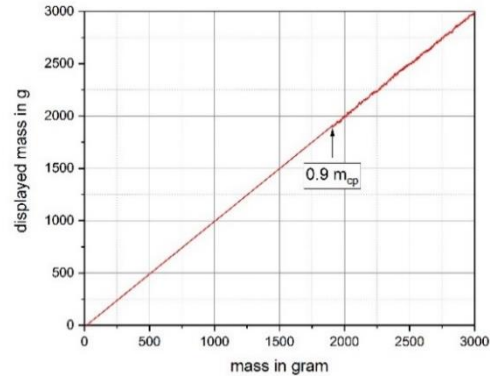
Figure 7 shows the tested two-interval load cell. In figures 8 and 9 selected results are presented.



**Fig. 7.** Two-interval load cell in testing machine



**Fig. 8.** Placing of single masses 1-10 gram



**Fig. 9.** Continuous loading up to 3kg, automatically range transition according equation (6)

## 6. Conclusions

The new developed two-interval load cell system allows the measurement of masses in the range 0 - 300kg. Up to 2kg the resolution is better than one gram. Range switching is stepless and automatic without operator intervention.

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# CONTROL OF A SPHERICAL PARALLEL MANIPULATOR WITH THREE DEGRESS OF FREEDOM

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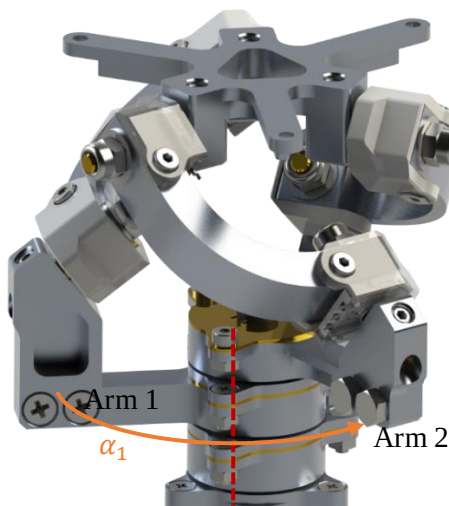
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## 1. Introduction

In robotic applications, it is often necessary to orient a sensor quickly. Spherical parallel manipulators (SPM) are well suited for this purpose since they offer superior dynamics and structural stiffness as compared to serial manipulators [1]. To control them, however, the kinematic equations have to be known. In this paper, a SPM with three degrees of freedom and the kinematic equations describing its mechanical properties are presented.

## 2. Hardware setup

Fig. 1 shows the spherical parallel manipulator this paper is dealing with. At the bottom there are three arms which can be rotated around the vertical axis. They are the only actuated links of the manipulator. In their normal position they draw an angle of  $120^\circ$  against each other, so that the sensor mount has the vertical axis of the manipulator as its normal vector.



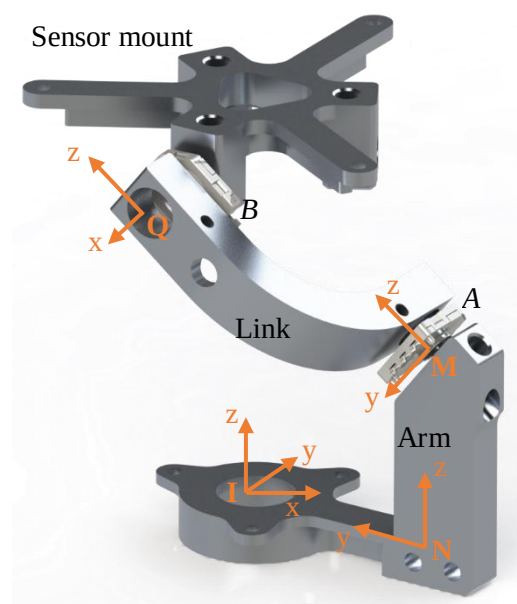
**Fig. 1.** Hardware setup of the spherical parallel manipulator. The dashed red line shows the vertical axis the three arms can be rotated about.

Three binary links connect the lower arms with the sensor mount. At their nodes the links are

connected via pivot joints allowing a rotational movement around one axis. For control purposes all six joints are equipped with analog potentiometers allowing for a measurement of the current angles of the system.

## 3. Kinematic equations

To derive the kinematic equations several coordinate systems are introduced. Each of the lower arms has a body-fixed coordinate system ( $N_{1,2,3}$ ) with the z-axis pointing up and the y-axis pointing towards the centre of rotation of the arm, as shown in Fig. 2. Since the joints connecting the links and the arms are inclined, the inclination angle ( $\theta = 52^\circ$ ) is encoded in the coordinate systems  $M_{1,2,3}$  which are fixed to the arms. Moreover, there are three coordinate systems  $Q_{1,2,3}$  which are fixed to the links. Their x-axes are aligned with the rotational axes of the upper pivot joints. The sensor mount has three body-fixed coordinate systems ( $P_{1,2,3}$ ) with their x-axes being coincident to the x-axes of  $Q$ , as shown in Fig. 3.



**Fig. 2.** One arm, link and the sensor mount of the SPM with coordinate systems to describe the kinematics.

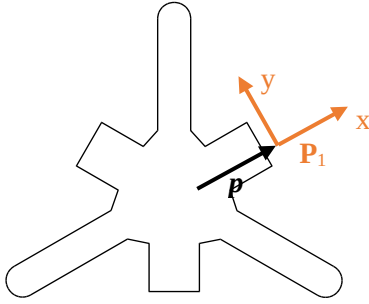


Fig. 3. Top view of the sensor mount.

Let  $R_x(\phi)$ ,  $R_y(\phi)$  and  $R_z(\phi)$  denote elementary rotations about the x-, y- and z-axis, respectively [2]. Further, let  $\alpha_1$  be the angle drawn between arm 1 and 2 and  $\alpha_2$  be the angle drawn between arm 1 and 3 (positive direction). The rotations from  $N_0$  to  $N_1$  and  $N_2$  describing the angular positions of the arms become:

$$R_{N_1 N_0} = R_z(\alpha_1) \quad (1)$$

$$R_{N_2 N_0} = R_z(\alpha_2) \quad (2)$$

A vector  $v$  given in the coordinate system  $N_0$  can then be rotated into the coordinate system  $N_1$ :

$${}_{N_1} v = R_{N_1 N_0} \cdot {}_{N_0} v \quad (3)$$

To transform  $N$  into the inclined coordinate systems of the arms,  $M$ , a rotation about the x-axis is applied (indices 1,2,3 are omitted for readability):

$$R_{MN} = R_x(\theta) \quad (4)$$

With  $\beta_{1,2,3}$  describing the angles between arms and links the rotations from  $M$  to the body-fixed coordinate systems of the links,  $Q$ , become:

$$R_{QM} = R_z(\beta) \quad (5)$$

Since the x-axes of the sensor mount coordinate systems  $P$  and the link coordinate systems  $Q$  coincide, the corresponding rotation is

$$R_{PQ} = R_x(\gamma) \quad (6)$$

with  $\gamma_{1,2,3}$  being the rotations about the upper pivot joints.

#### 4. Control approach

There are several ways to define the target orientation of the sensor mount. Possible options are, for example, Euler angles and unit quaternions. For the SPM, however, we decided to use the normal vector of the sensor mount with respect to an inertial system  $I$ , which is fixed to the mounting of the manipulator and has its z-axis coinciding with the vertical axis of the system, cf. Fig. 1 and Fig. 2.

From the normal vector  $n$ , setpoints for the arms,  $\alpha_{1,set}$  and  $\alpha_{2,set}$ , can be calculated. Unfortunately, the current position of the arms,  $\alpha_1$  and  $\alpha_2$ , cannot be measured directly due to constructional limitations. Therefore, we use the measurements from the analog potentiometers which provide  $\beta_{1,2,3}$  and  $\gamma_{1,2,3}$ .

Given the angles at the upper pivot joints, i.e.  $\gamma_{1,2,3}$ , the rotation of the sensor mount with respect to the fixed inertial system  $I$ ,  $R_{IP_1}$ , can be computed. From the geometry shown in Fig. 3 it follows that:

$$R_{IP_2} = R_z(120^\circ) \cdot R_{IP_1} \quad (7)$$

$$R_{IP_3} = R_z(240^\circ) \cdot R_{IP_1} \quad (8)$$

Let  $A_{1,2,3}$  denote the centre points of the pivots between the arms and links and let  $B_{1,2,3}$  denote the corresponding points between the links and the sensor mount, as shown in Fig. 2. Given a vector  ${}_P p$  describing the position of  $A_1$  in the  $P_1$  coordinate system, as shown in Fig. 3, and a vector  ${}_Q \overline{BA}$  that connects B and A in the  $Q$  system,  $B_{1,2,3}$  can be calculated as follows in the inertial system  $I$ :

$${}_I B = R_{IP} ({}_P p + R_x(\gamma) \cdot {}_Q \overline{BA}) \quad (9)$$

Then, the current angles of the arms can be computed:

$$\alpha = \tan^{-1} \frac{{}_I B_y}{{}_I B_x} \quad (10)$$

with  ${}_I B_x$  and  ${}_I B_y$  being the first and second component of the vector, respectively.

Eventually, to form a closed-loop control system we use a simple PID controller.

#### 5. Summary and Conclusion

The hardware setup of a spherical parallel manipulator with three degrees of freedom was presented and equations describing its kinematics were derived. In addition, a possible way to control the system was shown that uses indirect measurements of the control variables and a PID controller.

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# APPLICATION OF AN ELECTRIC CONTACT SENSOR IN MACHINING

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## 1. Introduction

Productivity of machining operations depends on the spindle speed, feed rate, and axial/radial immersions. Tuning of these parameters however does not necessarily provide better cutting conditions, since large amplitude vibrations, called chatter, can occur. Chatter spoils the surface quality, results in extensive noise, and increases toolwear. Detection and avoidance are therefore highly important tasks in machining [1]. The spectra of acquired signals provide information about the occurrence of chatter, but it cannot always be clearly recognized due to noise and measurement inaccuracies. Here, an electric contact-based device is presented in Fig. 1., that is free from environmental noise and only based on the physical contact of the tool and the workpiece.

## 2. Mechanical model of milling

The model of milling operation is presented in Fig. 1. The dynamical model is represented by

$$\ddot{\mathbf{r}}(t) + [2\zeta_k \omega_{n,k}] \dot{\mathbf{r}}(t) + [\omega_{n,k}^2] \mathbf{r}(t) = \mathbf{U}^T \mathbf{F}(t, \mathbf{r}, \mathbf{r}_\tau), \quad (1)$$

where  $\mathbf{r}(t)$  is the modal coordinate vector,  $\mathbf{U}$  is the mass-normalized modal transformation matrix,  $\zeta_k$  is the relative damping ratio, and  $\omega_{n,k}$  is the natural angular frequency. Let us introduce  $\mathbf{r}_\tau := \mathbf{r}(t - \tau)$  for brevity, where  $\tau = N\Omega/60$  is the time delay,  $N$  is the number of cutting teeth, and  $\Omega$  (rpm) is the spindle speed. The stability properties of (1) are determined by the Floquet multipliers of the associated variational system [2]

$$\ddot{\mathbf{u}}(t) + [2\zeta_k \omega_{n,k}] \dot{\mathbf{u}}(t) + [\omega_{n,k}^2] \mathbf{u}(t) = \mathbf{U}^T \mathbf{G}(t) \mathbf{U} (\mathbf{u}(t - \tau) - \mathbf{u}(t)), \quad (2)$$

where  $\mathbf{u}(t) = \mathbf{r}(t) - \mathbf{r}_p(t)$  is a small perturbation about the  $\tau$ -periodic solution  $\mathbf{r}_p(t)$  of (1), and  $\mathbf{G}(t)$  is the  $\tau$ -periodic directional matrix. Stability analysis of (2) can be performed by numerical methods, such as the semi-discretization technique [2] or the multi-frequency solution [3].

## 3. Chatter identification in practice

The stationary solution of (1) under stable machining parameters is periodic with  $\tau$  and the corresponding spectrum contains dominant peaks only at the integer multiples of the tooth-passing frequency  $f_{\text{Tooth}} = 1/\tau$ . However, when the periodic solution loses stability, the frequency of the increasing vibrations will differ from the tooth-passing frequency. The harmonics of the arising quasi-periodic vibrations can be observed on the spectra of the sensors used to monitor the operation.

In most of the cases the signals are collected by dynamometers, accelerometers and industrial microphones. Chatter is typically identified after the operations, when data are evaluated. Also, very often the visual observation of the machined surface after the cutting process helps to ascertain chatter. All of the above mentioned data acquisition methods have their drawbacks, such as

- the microphone typically contains many external noise from the environment;
- the acceleration sensor cannot be placed onto the rotating tool, only a further mounting point can be used, where the sensor measures the acceleration through bearings and other moving elements;
- force sensor is usually placed under the workpiece: in case of a flexible structures the measured force is not identical to the theoretical cutting force.

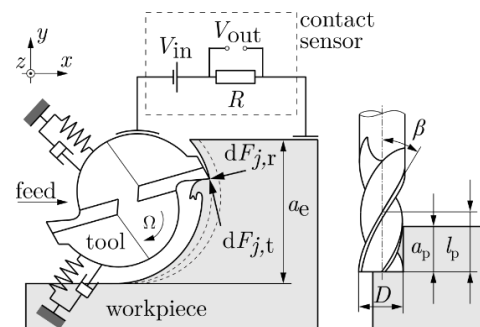


Fig. 1. Dynamical model and the electric contact sensor.

In case of chatter-free machining, the signal of the electric contact sensor is presented in Fig. 2(a).

The voltage drops to zero if there is no physical contact between the tool and the workpiece. Fig. 2(b) presents the spectrum of the signal of the contact sensor. Since the operation was stable, dominant peaks should appear only at the tooth-passing frequency and its integer multiples (dashed vertical lines). Due to the runout effect, smaller peaks can be seen halfway between the multipliers of the tooth-passing frequency (dotted line), but it is still not identified as chatter. Fig. 2(c) shows the spectrum of the microphone, which contains irrelevant frequency components that might correspond to the vibration of the fan of the cooling system or other devices.

In case of chatter (Fig. 3), the tool either flies over the material or contacts the top surface of the workpiece during the noncutting period due to large amplitude aperiodic vibrations. This chaotic contact can be seen in Fig. 3(a). The spectrum of the contact sensor in Fig. 3(b) identifies clearly the frequency components (red crosses) of the chatter vibration. Meantime, Fig. 3(c) indicates the spectrum of the microphone that identifies the most dominant chatter frequency at  $\sim 800$  Hz. In this configuration the microphone was also found to be suitable for chatter identification, however, the spectrum still contains several additional fake peaks which make the chatter detection uncertain.

**Table 1.** Machining parameters

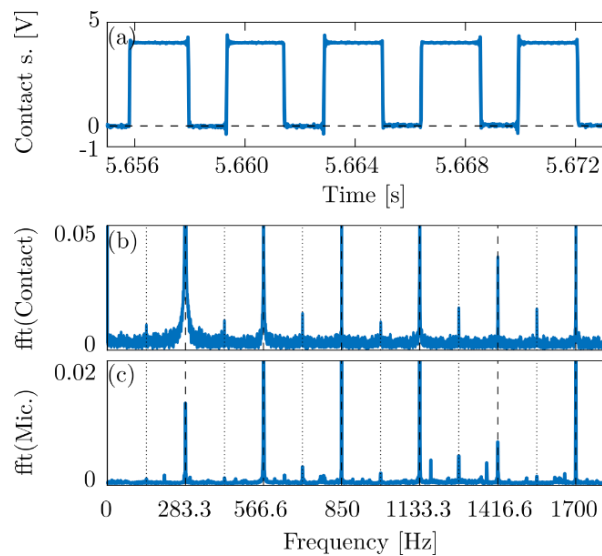
|                  |                                                            |
|------------------|------------------------------------------------------------|
| <b>Tool</b>      | Tivoly P615H, $N = 2$ , $D = 16$ mm,<br>$\beta = 35^\circ$ |
| <b>Workpiece</b> | Aluminum 2024-T351                                         |
| <b>Process</b>   | Down-milling, $a_e = 8$ mm,<br>$f_z = 0.1$ mm/teeth        |

## 4. Conclusions

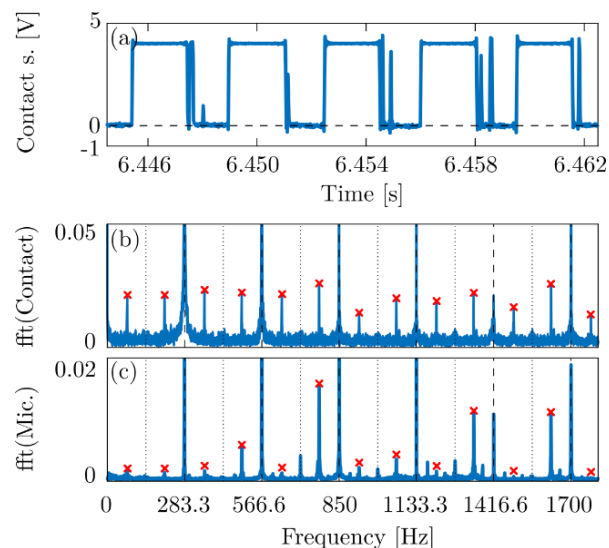
An electric contact sensor was developed and tested to detect chatter during machining operations. This low-cost instrument is able to identify chatter frequencies and is not affected by environmental noise. Therefore, it provides an alternative and economical way of chatter identification when traditional instruments cannot be applied.

## Acknowledgements

The research leading to these results has received funding from the European Research Council under the European Unions Seventh Framework Programme (FP/2007-2013) / ERC Advanced Grant Agreement n. 340889 and it was supported by the Hungarian Scientific Research Fund (OTKA FK-124462 and PD-124646).



**Fig. 2.** Stable operation (chatter-free)  
( $\Omega = 8500$  rpm,  $a_p = 3$  mm).



**Fig. 3.** Unstable operation (chatter)  
( $\Omega = 8500$  rpm,  $a_p = 4$  mm).

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# COMPARISON OF COOLING SIMULATIONS OF INJECTION MOULDING TOOLS CREATED WITH CUTTING MACHINING AND ADDITIVE MANUFACTURING

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## 1. Introduction

Nowadays injection molded products are gaining ground worldwide. The reason of this is that their production time is relatively short, and during the lifetime of a tool it can even produce hundreds of thousands of products. The manufacturing speed depends on the injection molding cycle, which is one of the most time-consuming parts of the cooling time [1]. As a consequence, the primary aim of the manufacturers is to minimize the cooling time within the injection molding cycle. We investigated whether the cycle time of an injection molding can be reduced by using an additive manufacturing instead of traditional machining.

## 2. Preparations

When selecting the product to be examined, some important aspects had to be set up. Since it is not our intention to completely replace conventional injection molding tools, it is not recommended to choose a simple product geometry. However, the simulation time would considerably increase if we chose a complicated and large product [2]. The geometry of the chosen product is not so complicated, but it has a sufficient depth to make using of a deep tool insert. Additionally, the technology sheet of the product was available, the simulations are compared with the real production conditions.

The simulation of the process was created with the tool and the cooling circuit of the selected product. After that the cooling circuits of the tool inserts have been modified so, that those can not be formed by machining anymore, only by additive manufacturing. Simulations were also made of the new cooling circuit thus formed.

## 3. Simulations

The simulations were run both in standard and in transient mode. Because of the transient simulation has stopped after two cycles, the standard run resulted the same realistic picture of the process. Comparison of standard and transient results also confirmed this [3].

During the simulations, the temperature of the product and the tool, the degree of warping of the product were examined. The cooling time is determined: if 95% of the elements reached the ejection temperature was given on the technology sheet, the product can be removed safely.

## 4. Results

According to the collected data, it is possible to manufacture injection moulding tools by additive manufacturing. Using this technology, 100% compactness is available [3], [4]. After a proper heat and surface treatment, the mechanical properties of the tools will be even to the tools created by cutting machining.

Based on the results obtained, the 21 second cooling time originally dropped by one second, representing a nearly five percent improvement of cooling time. Using the previous experiences on the cooling circuit, the injection molding tool was further upgraded. Thus, the required cooling time decreased to 19 seconds.

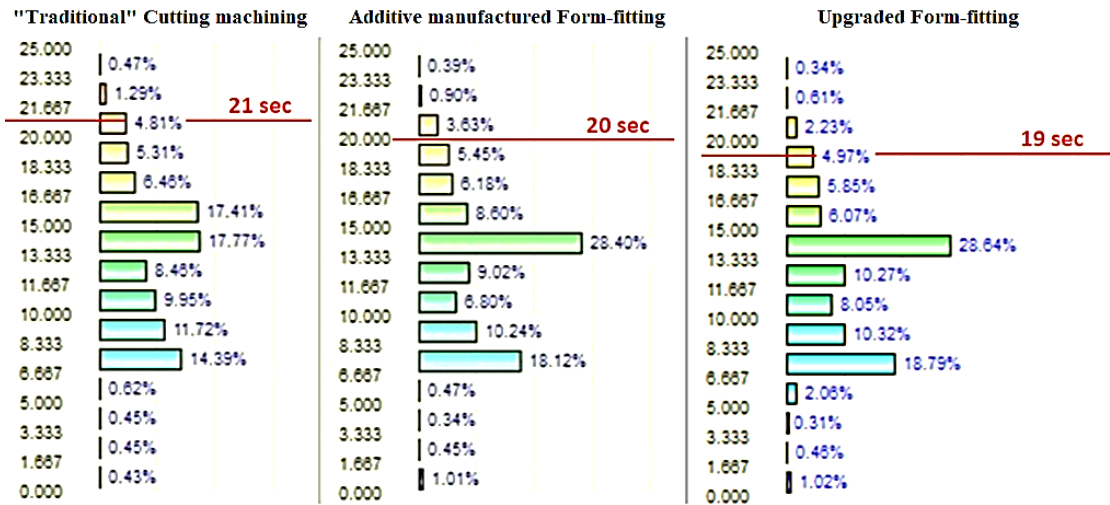


Fig. 1. Cooling time of the elements until they reach the ejection temperature.

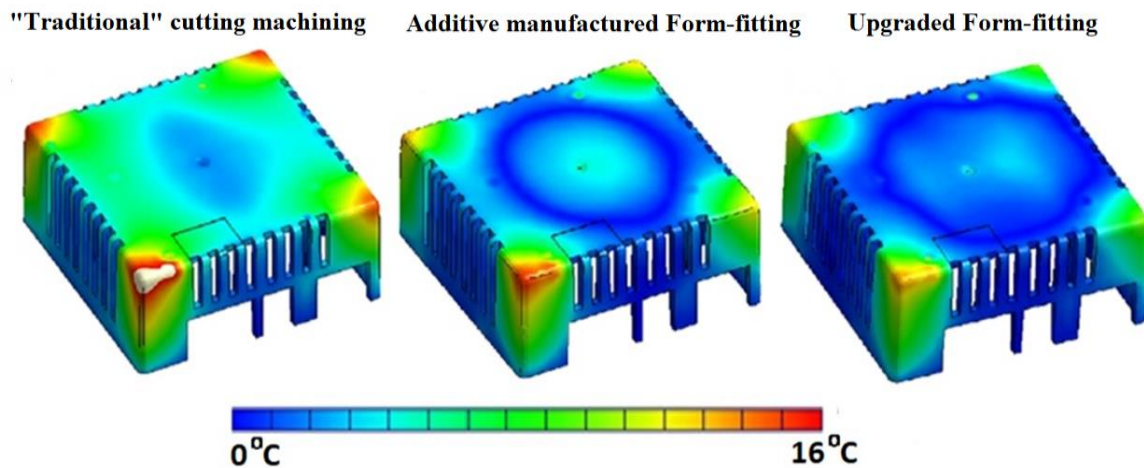


Fig. 2. Thermal difference between inside and outside surface of the product at the moment of ejection, which is proportional to the degree of warping.

## 5. Summary

The following conclusions can be made:

- Injection moulding tool created by additive manufacturing is possible and has even attributes as tools created by cutting machining.
- In the case of simple product geometry, the use of additive manufacturing tools is not necessary
- In the manufacture of high-complexity products, the form-fitting cooling system (which is not possible with traditional manufacturing methods) can greatly increase productivity, due to more efficient heat shift and shorter cycle time.

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# CUTTING FORCE MEASUREMENT FROM ACCELERATION SENSOR IN MILLING OPERATION

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## 1. Introduction

In the manufacturing industry, productivity is a key factor besides quality, efficiency and sustainability. However, productivity cannot be increased arbitrarily due to undesired vibration that may arise during the cutting process [1]. The prediction and avoidance of these vibrations are still an active researched area.

In order to thoroughly discover and understand the dynamic behavior of machining processes, an accurate and reliable identification of the resultant cutting force is an important factor. In general, it can be measured accurately by means of dynamometer, although, these sensors are usually quite expensive measurement equipments. In addition, for sufficiently accurate measurement results, it may be necessary to take into account the dynamic characteristics of dynamometers which limits the reliable bandwidth of the measurement.

The purpose of this paper is to propose an alternative method that makes it possible to predict cutting force without using expensive dynamometer. First, the basic idea and the developed method of the alternative cutting force measurement is presented, in which the force is predicted based on measured acceleration signal treated with inverse Frequency Response Function (FRF). This section gives the mathematical background for the identification of the cutting force. Then, this measurement technique is applied on a case study, which is the main contribution of the paper (see Section 3). Finally, in order to validate the proposed method, the reconstructed result is compared with cutting force measurement with dynamometer.

## 2. Reconstruction of cutting force

In order to reconstruct the cutting force without dynamometer, the key idea is to capture the system's response and using the FRF, which describes the dynamic behavior of the system [2]. The FRF, also called transfer function, describes the

ratio between the output (resultant vibration  $x(t)$ ) and the input (cutting force  $F(t)$  in frequency domain. It is usually determined by modal analysis, typically by means of impact test with modal hammer [2]. Once the FRF is known, then the resultant cutting force  $F(t)$  can be reconstructed as follows:

$$F(t) = \mathcal{F}^{-1} \left\{ H^{-1}(\omega) \mathcal{F} \{ x(t) \} \right\}, \quad (1)$$

where  $H(\omega)$  is the measured FRF, while  $\mathcal{F}\{\square\}$  and  $\mathcal{F}^{-1}\{\square\}$  denote the Fourier and inverse Fourier transformation, respectively.

### 2.1 Sampled discrete case

In practice, the FRF and the response are measured as sampled discrete data, therefore, the proposed method can be applied in discrete form as the following steps shows:

$$x(t) \xrightarrow{\text{FFT}} \xi(\omega) \xrightarrow{H^{-1}(\omega)} \Phi(\omega) \xrightarrow{\text{IFFT}} F(t), \quad (1)$$

where  $\xi(\omega)$  and  $\Phi(\omega)$  are the FFT of  $x(t)$  and  $F(t)$ , respectively. The number of the measured discrete values during one tooth-passing period  $m_x$  and the number of the reconstructed cutting force values  $m_F$  can be given as:

$$m_x = \text{floor}(2f_s T), \quad m_F = \text{floor}(2f_{\max} T), \quad (1)$$

where  $f_s$  is the sampling frequency of the acceleration measurement, while  $f_{\max}$  is the available bandwidth of the mechanical system's FRF. The time period  $T$  in seconds reads as:

$$T = 60/(nN), \quad (1)$$

where  $n$  is the spindle speed in rpm and  $N$  is the number of cutting teeth.

Note that the number of the measured points of the periodic vibration signal is usually larger than the number of the reconstructed cutting force points, since the bandwidth  $f_s$  is usually greater than the relevant bandwidth  $f_{\max}$  of the measured FRF.

Consequently, the reconstructed cutting force is more accurate in lower spindle speed domain, where sufficiently large number of reconstructed points are available.

### 3. Case study

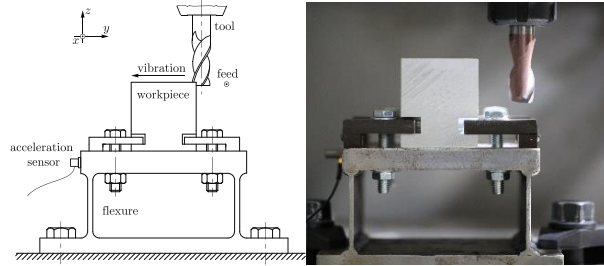
The above described measurement technique is applied in a case study during a peripheral milling test along straight path with down-milling operation. In the experimental procedure, the workpiece (Aluminum 2024-T351) is clamped onto the top of a flexible structure, which was designed to mimic the dynamics of a single-degree-of-freedom system (SDoF) (see Fig. 1) [3]. The structure is flexible only along  $y$  direction and can be considered to be rigid in the other directions. The measured FRF can be seen in Fig. 2c.

The cutting tests was performed with spindle speed  $n = 4650$  rpm, radial immersion  $a_e = 2$  mm, feed per tooth  $f_z = 0.05$  mm/tooth and axial immersion  $a_p = 2$  mm. The TIVOLY P615H endmill had diameter  $D = 16$  mm, number of flutes  $N = 2$ , helix angle  $\beta = 30^\circ$  and rake angle  $\kappa = 90^\circ$ . The response was acquired by NI cDAQ-9178 Chassis with NI 9234 Module at 51.2 kHz sampling rate and PCB 352C23 type acceleration sensor. One time period  $x(t)$  and its FFT can be seen in Fig. 2a and 2b, respectively. Reconstructed cutting force is plotted in Fig. 2d together with direct force measurement with Kistler dynamometer (9129AA) repeated with the same technological parameters. Note that the difference is in the range of the typical error of force measurement with dynamometers.

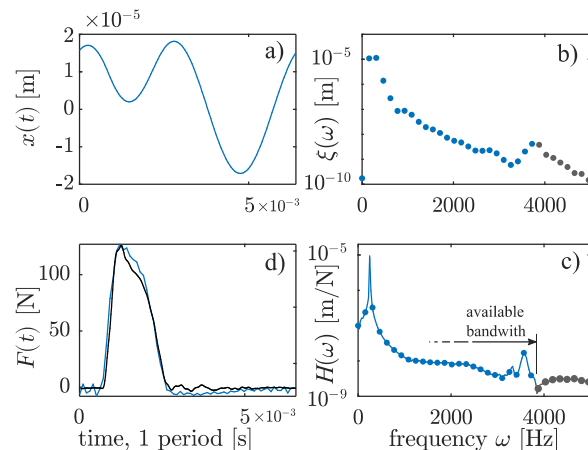
### 4. Conclusions

In this paper, an alternative method is proposed which uses only accelerometer sensor and modal hammer. It is capable to measure the resultant cutting force during milling operations directly from vibration measurements without using expensive dynamometer. The method is based on the followings: the response of the system is captured by acceleration signal, from which the resultant cutting force can be reconstructed by means of the multiplication of the inverse FRF.

The efficiency and accuracy of the proposed methodology were validated experimentally in laboratory environment by means of direct cutting force measurement with dynamometer.



**Fig. 1.** Schematic figure and experimental setup for milling with single-degree-of-freedom experimental structure [3].



**Fig. 2.** a) Periodic response during one time period; b) FFT of the periodic response; c) measured FRF (curve) and interpolated values (dots); d) cutting force reconstructed by acceleration sensor (blue) and measured by means of dynamometer (black).

### Acknowledgements

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# IMPROVED SPINDLE SPEED MEASUREMENT BY ANALOGUE SENSOR

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## 1. Introduction

The demand for higher productivity in milling operation leads to high spindle speed, feed rate, and axial/radial immersion. These technological parameters are usually limited by harmful vibrations. One type these vibrations are generated by the periodical cutting forces. These forced vibrations can have undesired large amplitude at the resonant spindle speeds [1]. The other type of vibrations are the so-called chatter vibrations related to the surface regeneration effect [2]. These self-excited vibrations emerge from the unstable forced periodic orbits and they are very harmful to the milling tool and leads to unacceptable surface quality.

The stable forced vibrations is periodic with  $\tau = Nn$ , where  $N$  is the number of cutting teeth, and  $n$  is the spindle speed and the corresponding spectrum contains dominant peaks only at the integer multiples of the tooth-passing frequency  $f_{\text{Tooth}} = 1/\tau$ . In contrary, the chatter vibration has a different so-called chatter frequency which is usually close to the dominant natural frequency of the system.

In practice it is very important to identify the periodic terms to improve the analysis of the periodic components (e.g.: in the measurement of the cutting force characteristic or in the analysis of the resonant vibrations) and also to be able identify the unstable chatter vibration.

The periodicity of the milling process is defined by the spindle speed, which must be measured with high accuracy. In the present work, an improved measurement is presented which is capable to provide higher precision than the sampling rate.

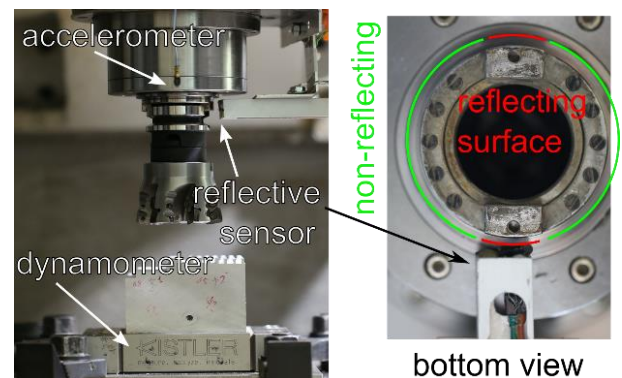
## 2. Spindle speed measurements

The CNC machine tools have a built-in spindle speed measurement system, which is used for the control, however, the output of these sensors are usually not available. The user can only define the

desired spindle speed, however, the speed fluctuates around it, furthermore, it can have a constant offset error, too. If the periodic component would be sectioned based on the desired spindle speed, a really slight error (<0.1%) could lead to a false results under a few seconds at a high spindle speed, due to the accumulating phase shift difference.

An additional problem of the external sensors is that the rotating spindle can be reached only in a small area due to the spindle housing, in addition, the sensors should not be placed close to the cutting area due to the flying chips and the cutting fluids.

We present here a simple measuring unit based on an reflective phototransistor (see Fig.1), which provides a trigger sign for the sectioning.

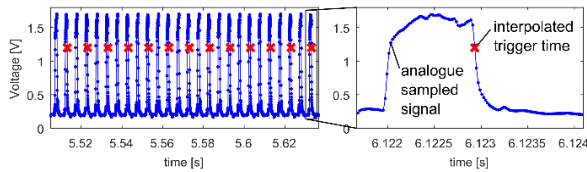


**Fig. 1.** Reflective switch sensor for the spindle speed measurement together with the other typical sensors (accelerometer, dynamometer and microphone).

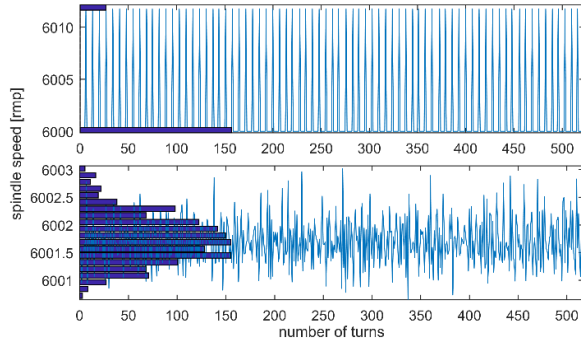
## 3. Effect of the sampling time on digital signal

Some of these type of reflective sensors has built-in *ambient light rejection* to compensate the lights from the environment and have a digital output. Although, the internal electronics of these sensors have a low sampling rate (bandwidth). The angle accuracy of these the digital signal under one revolution is:

$$\Delta\alpha = 360^\circ \frac{n}{f_s} \quad (1)$$



**Fig. 2.** Continuous change of the analogue signal and the linearly interpolated trigger time.



**Fig. 3.** Distribution of the spindle speed for digital signal and for the interpolated analogue signal at 51.2kHz sampling rate.

For example in case of the Optek / TT Electronics OPB720 reflective switch ( $f_s=7$  kHz) at  $n=30\,000$  rpm the angle resolution is only  $\sim 25.7^\circ$ . To reach a desired  $0.1^\circ$  accuracy 1.8 MHz sampling rate would be needed, which could be achieved without internal logic only, (e.g. Vishay CNY70 reflective sensor).

#### 4. Improved prediction with analogue signal

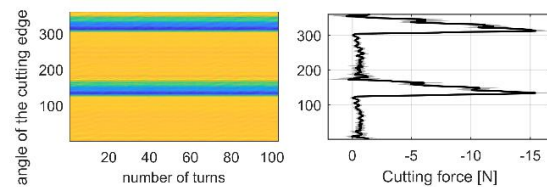
In a typical measurement, the signal of the spindle speed sensor is used together with other analogue signals e.g.: accelerometer or force signals of a dynamometer (see Fig. 1), which typically leads to limited bandwidth. In case of the widely used National Instrument Compact DAQ Chassis cDAQ-9178 and NI-9234 module, the sampling rate  $f_s=51.2$  kHz, leading to 102.4 samples per turn and  $\Delta\alpha = 3.51^\circ$  angle accuracy at  $n=30\,000$  rpm. In case of a digital signal the trigger will arrive after 102 or 103 samples, with the resulting spindle speed 29 825 rpm and 30 118 rpm, respectively.

Although, the Vishay CNY70 sensor can also be used as an analogue signal, which creates a fast, but continuous signal change (see Fig. 2). Increased accuracy can be achieved if a linear interpolation is used. Figure 3 shows the distribution of the spindle speed (computed based on the time difference of the trigger signals) for digital and interpolated analogue signals at  $n=6\,000$  rpm. It is clearly seen that the digital signal leads to two distinct value belongs to two neighboring integer multiplier of the sampling

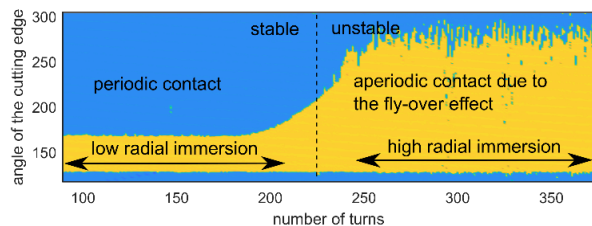
time. In contrast, the interpolated signal is capable to capture the Gaussian fluctuation of the spindle speed together with the small offset error  $\sim 1.69$  rpm. In case of a digital signal, the presented Gaussian distribution could be captured only above sampling rate  $f_s > 760$  KHz and with  $\sim 0.05^\circ$  angle accuracy even at the moderate  $n=6000$  rpm spindle speed.

#### 5. Sectioning of the signals

The achieved accuracy of the interpolated analogue trigger signal can be used for sectioning the time signal. This is can be used for proper averaging the periodic signal (see Fig. 4) and to detect the aperiodic chatter vibration (see Fig. 5).



**Fig. 4.** Time signal and its sectioned version, together with the properly averaged value.



**Fig. 5.** Sectioned contact sensor and the detected chatter

#### 6. Conclusion

The proposed method: the linear interpolation of the analogue signal of the reflective sensor can be used in milling processes to improve the accuracy of the measurements, when the other analogue signals already limits the available bandwidths.

#### Acknowledgements

The research leading to these results was supported by the Hungarian Scientific Research Fund (OTKA FK-124462 and PD-124646).

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# DESIGN THINKING IN NUMERICAL AND EXPERIMENTAL EVALUATION OF THE STRUCTURE OF CONSTRUCTION ROBOT

**Damian DERLUKIEWICZ, Jakub ANDRUSZKO**

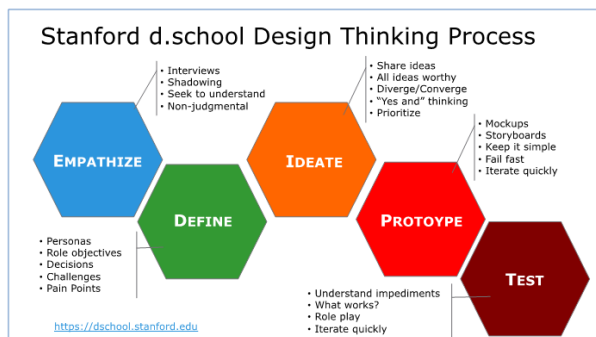
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## 1. Introduction

The Design Thinking is a five step methodology for searching and solving the problem that can result in an innovative solution. Design Thinking process at the beginning defines the core problem and then by the different tools helps to find the solutions, always with the needs of the user [1].

The main goal of the Design Thinking process is to develop solutions that are both cost-effective in business, technically feasible and desirable by clients (responding to real needs). Its essence consists in the systematic creation and implementation of innovative solutions. Through innovative solutions, we understand not only new products, but also innovative technologies, services, strategies, processes, business models and educational programs. Regardless of the activity, whether it is scientific, business, industrial, educational or administrative, the Design Thinking methodology supports the development of innovation and guarantees the creation of non-standard solutions.

Basically Design thinking consists of five key elements, Fig. 1.



**Fig. 1.** Design Thinking Process [3]

This systematic action, thanks to its universal character, has a wide application, wherever we face

problems that do not have one obvious solution or a rigid framework. Design Thinking is first of all a methodology focused on people as users, which through interdisciplinary gives the opportunity to plan, test and implement innovations in products and services of the organization in the conditions of their continuous development.

These five steps are [1,2]:

- **EMPATHIZE:** Work to fully understand the experience of the user for whom you are designing. Done by the observation, interaction, and immersing yourself in their experiences.
- **DEFINE:** Process and synthesize the findings from your empathy work in order to form a user point of view that you will address with your design.
- **IDEATE:** Explore a wide variety of possible solutions through generating a large quantity of diverse possible solutions, allowing you to step beyond the obvious and explore a range of ideas. Performing the brainstorming process
- **PROTOTYPE:** Transform your ideas into a physical form so that you can experience and interact with them and, in the process, learn and develop more empathy.
- **TEST:** Try out high-resolution products and use observations and feedback to refine prototypes, learn more about the user, and refine your original point of view.

## 2. Numerical evaluation of construction robot structure

For the proper assumption of the boundary conditions in the numerical simulation the Empathy stage from Design Thinking methodology was

conducted. The deep interview with the operator enables to define the positioning of the arm that operator is setting during the demolition working with the robot. That helps to understand why it differs from the proper settings according to manual or proper as far as strength of the structure is concerned operation. The one of the numerical simulation results in the specific position of the arm that was set by the one of operators is presented in figure 2. In that case the stresses are fortunately at a low level but it shows that the designer of the robot needs to forecast all the possible settings of arms locations.

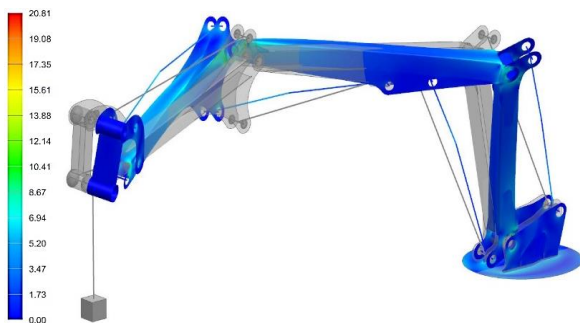


Fig. 2. Example of the stresses level in the working arm structure of construction robot

### 3. Experimental evaluation of construction robot structure

Based on the empathy stage of DT conducted to estimate the different boundary conditions for numerical calculations the experimental testing was also conducted to validate the numerical simulations results. The operator during testing set the positioning of the arm system in accordance to the boundary conditions elaborated from the Design Thinking Empathy stage.

The experimental testing with use of strain gages and accelerometers was conducted to validate the results obtained during the numerical simulation as well to observe the state of the machine during different but often positioning of robot arm system. The experimental testing of the robot is shown in Fig. 3.

The experimental testing also confirmed the results of operators deep interview conducted at the Empathy stage, what shows that operators of that kind of robots do not feel the situation, when the robot is working in improper conditions (high vibrations) because they are located always a few steps far from the machine, with the remote control

system, not as in classical construction machines sitting on it.



Fig. 3. Example of experimental testing during operation of construction robot

### 4. Results

The Design Thinking methodology was used to elaborate the proper and unexpected boundary conditions of operation of construction robot in order to evaluate the strength level of the robot at that conditions. The numerical simulations evaluate the stress level of the robot in defined positions of the arm system structure, where the experimental testing confirmed the state of strength and validate the numerical simulations.

Carrying out the DT Empathy stage can be very useful to estimate the different unexpected conditions of machines during operation.

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# EXPERIMENTAL AND NUMERICAL EVALUATION OF WAVE IMPACT STRESS ON A COMPOSITE BOAT HULL

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## 1. Introduction

Composite materials are used in the boatbuilding industry for a wide range of advantages, both economical and performance related. As in any vehicle construction, safety concerns must be taken into account, therefore the structural calculations are done using the scantling rules of the registers of shipping.

High speed crafts are subjected to wave impact at cruising speed. The register rules use statistical data for evaluating pressure and acceleration along the hull for this extreme loading condition. Sometimes the evaluation underestimates the real loading condition measured experimentally [1], [2], [3].

This paper will use experimental data recorded on a full scale instrumented boat hull to compare the stress on various measuring points with the stress obtained in a numerical simulation using FEM. Pressure loads applied to the numerical model are calculated according to the rules published by DNV-GL [4].

## 2. Materials and methodology

The composite material used in this research is hand lay up fiberglass in polyester resin matrix, single skin. In order to ensure best accuracy with numerical results, samples of the composite material were extracted from a hull excerpt and tested in the laboratory (Fig. 1).



Fig. 1. Hull excerpt (left) and sample testing (right)

Numerical simulations were conducted on a global model of the boat used in the experiment, based on the DNV-GL pressure loads for slamming impact on a wave crest and wave hollow (Fig. 3)

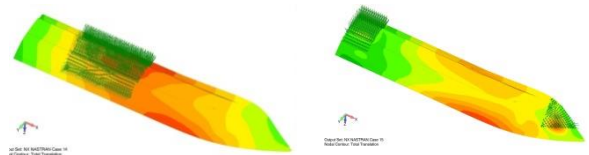


Fig. 2. Pressure load on structure: crest landing (left) and hollow landing (right)

The experiment was conducted on a full scale composites boat equipped with 10 strain gauges and 2 accelerometers. Data acquisition was done at a frequency of 50hz on 2040 readings blocks. The speed of the boat was kept constant at 6 m/s using a GPS device. The location of the measuring devices is shown in Figure 3, where 1 to 10 are the strain gauges and AccPp and AccPv are the accelerometers. Using a linear interpolation the value of the acceleration in the center of mass can be approximated based on the readings of AccPv and AccPp.

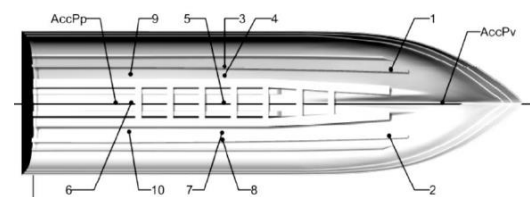


Fig. 3. Instrument location in the hull

The register rules offer a methodology for acceleration calculation in the center of mass, based on the hull geometry particulars and speed. For 6 m/s the value of the acceleration calculated based on the before mentioned rules in the center of mass is 25.70 m/s<sup>2</sup>.

### 3. Results

In Table 1 there is shown an excerpt of the measurement record where the value of the acceleration matches the calculated value based on the rules.

Table 2 shows the stress results of the numerical simulation using FEM on the location of the strain gauges, for both crest landing and hollow landing. FEM model used an orthotropic 2D material model.

**Table 1.** Experimental readings.

| Time sampling [s]       |       | 31.98   | 32      | 32.02   |
|-------------------------|-------|---------|---------|---------|
| Normal stress [MPa]     | M1    | 0.031   | 0.073   | 0.053   |
|                         | M2    | 0.071   | 0.155   | 0.119   |
|                         | M3    | 0.051   | 0.042   | -0.031  |
|                         | M4    | 0.219   | 0.107   | -0.092  |
|                         | M5    | -0.012  | 0.026   | 0.089   |
|                         | M6    | -0.033  | -0.02   | 0.017   |
|                         | M7    | 0.217   | 0.112   | -0.054  |
|                         | M8    | 0.029   | -0.062  | -0.042  |
|                         | M9    | 0.003   | -0.008  | 0.021   |
|                         | M10   | 0.002   | 0.01    | 0.027   |
| Acc [m/s <sup>2</sup> ] | AccPv | -31.542 | -31.542 | -31.542 |
|                         | AccPp | 2.317   | -0.036  | 0.079   |
|                         | acc   | -24.402 | -24.898 | -24.873 |

**Table 2.** Numerical results.

|                     |     | Crest   | Hollow  |
|---------------------|-----|---------|---------|
| Normal stress [MPa] | M1  | -0.0254 | 0.0429  |
|                     | M2  | -0.0278 | 0.0437  |
|                     | M3  | 0.6873  | -0.0417 |
|                     | M4  | -3.7063 | 0.3073  |
|                     | M5  | -3.6556 | 0.6825  |
|                     | M6  | -1.9155 | 0.2893  |
|                     | M7  | -3.7076 | 0.3073  |
|                     | M8  | 0.6778  | -0.0417 |
|                     | M9  | -1.8254 | 0.373   |
|                     | M10 | -1.8824 | 0.3688  |

### 4. Conclusions

Comparison of the results shows a better match for the values of the hollow landing case, however, not all the values are close to the numerical results. The reason of the inconsistency is that experimental impact is a combined hollow landing case, where the front end of the boat experienced a wave impact,

while the rear end kept contact with water. The numerical loading case conducted using the rules is an idealized situation, where both ends experience impact with waves while the middle of the hull is suspended.

Highlighted cells show the values of the readings that are similar to the numerical results. The range of error is between 0.72% and 29.38%.

Considering the numerical simulation as quasistatic and the experiment as a dynamic loading case, the range of error measured shows a reasonable fair match.

The loading cases as proposed by the register of shipping are properly formulated and offer a fairly realistic loading case. Extended experimental data can contribute to the adjustment of the rules as the improvement of the measuring devices leads to better accuracy.

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# OPTICAL STUDY FOR SPRINGBACK PREDICTION, THICKNESS REDUCTION AND FORCES VARIATIONS ON SINGLE POINT INCREMENTAL FORMING

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## 1. Introduction

The single point incremental forming (SPIF) process is a forming process that has gained an outstanding importance in the last few years, especially for the small batch and unique parts production, due to the advantages it offers. Despite the fact that this forming process appeared 25-30 years ago, the single point incremental forming is not yet analysed enough because of the multiple technological forming variants.

This forming process consists in movement of a punch on a projected trajectory with the purpose of locally forming the part. The formed part is obtained as a result of convoluting successive positions of the punch. Hence results the main advantage of single point incremental forming: reducing the forming forces relative to other conventional forming process. Another important advantage consists in the simple geometry of the forming tools. Practically with the same punch and die we can form a large variety of parts.

## 2. Materials and methodology

The main SPIF parameters of interest are the thickness reduction variation, the variation of the process forces and the precision of the obtained parts. The experimental testing setup is composed of the anthropomorphic robot (KUKA KR210), of a support that contains the die and of the optical system. The punch is fastened in a mandrel mounted on the robotic arm. Between the mandrel and the robotic arm the force transducer is fixed. The die has a single part, the lower one, and is placed in vertical position. It consists of a base plate, two supports, and the blankholder. The blank is stiffly fastened between the die and the blankholder by means of four screws on each side.

The die used to produce all parts was a rectangular plate with the active side dimension  $Ld = 150$  mm. The radius of the die was  $R = 3$  mm. The

selected geometry of the part, realized by single point incremental forming, is a pyramidal frustum. The punch follows a spiral trajectory. The first length of the trajectory followed by the punch in both directions was  $Lt = 40$  mm. The blank sheets used were individual blanks of square shape, with dimensions of 250 mm x 250 mm, made of deep-drawing sheet (DC04) and of initial thicknesses  $t = 0.4$  mm. In order to decrease the friction on the surface that came into contact with the punch, a thin layer of lubricant was applied.



Fig. 1. Experimental layout for researches on single point incremental forming process

For determining the forces, thickness reduction and precision and the influence of the technological parameters on these variables, we elaborated a full factorial experiment of  $2^2$  types, in which each of the parameters was varied on two variation levels: for the step on vertical direction ( $sz$ ) 0.1 mm, and 0.5 mm and for the punch diameter ( $dp$ ): 6 mm and 10 mm. Hence results a number of four cases, presented in Table 1.

For the force measurements the Quantum X data acquisition system was used. The optical system used for determining the thickness reduction and precision was an Aramis optical system, manufactured by Gom. The measuring technique consisted of acquiring an image at different acquisition rates.

**Table 1.** The four analysed cases

| Case no. | Vertical step [mm] | Punch diameter [mm] |
|----------|--------------------|---------------------|
| 1        | 0.1                | 6                   |
| 2        | 0.5                | 6                   |
| 3        | 0.1                | 10                  |
| 4        | 0.5                | 10                  |

It should be noticed that the Aramis computational system allows determining coordinates, displacements, strains and thickness reductions for each acquired image. From all of these, the authors chose to achieve in this paper the thickness reduction and two STL files of the part after the end of the forming process and after the exit of the punch (after the springback process). These two STL files were compared with the „theoretical” surface of the part and the errors in few distinguished points were determined.

### 3. Results

The results of the experimental researches were focused on determining the maximum values of  $\epsilon$  major strain, minor strain, thickness reduction, forces on z direction, forces on x and y directions. Table 2 presents the results for major strain, minor strain and thickness reduction at the end of the forming process.

**Table 2.** The results for major strain, minor strain and thickness reduction on SPIF process

| Case no. | Major strain [mm/mm] | Minor strain [mm/mm] | Thickness reduction [mm/mm] |
|----------|----------------------|----------------------|-----------------------------|
| 1        | 0.358                | 0.088                | 0.375                       |
| 2        | 0.382                | 0.101                | 0.401                       |
| 3        | 0.317                | 0.118                | 0.322                       |
| 4        | 0.349                | 0.126                | 0.353                       |

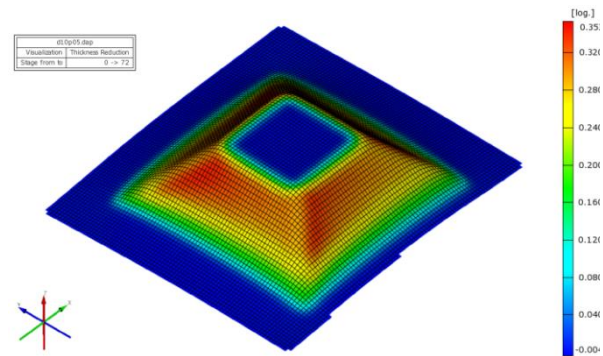
Table 3 presents the results for force on z, x and y direction.

**Table 3.** The results for forces on z, x and y direction on SPIF process

| Case no. | Fz [N] | Fx [N] | Fy [N] |
|----------|--------|--------|--------|
| 1        | 484    | 210    | 213    |
| 2        | 610    | 221    | 222    |
| 3        | 517    | 240    | 245    |
| 4        | 733    | 262    | 270    |

Figure 2 presents the thickness reduction variations for the case no. 4.

Table 4 presents the results for maximum value of the springback on the parts formed by single point incremental forming.



**Fig. 2.** Thickness reduction variation for the case no. 4

**Table 4.** The results for springback on SPIF process

| Case no. | Maximum value of springback [mm] |
|----------|----------------------------------|
| 1        | 0.761                            |
| 2        | 0.947                            |
| 3        | 0.775                            |
| 4        | 0.991                            |

### 4. Conclusions

Referring to the major strain and thickness reduction, these increase by increasing the vertical step and decrease by increasing the punch diameter. The force on vertical direction ( $F_z$ ) grows with the increasing of the vertical step as well as with the increasing of the punch diameter, while forces on horizontal direction ( $F_x$  and  $F_y$ ) are just a little influenced by them. The springback increases with the vertical step growth and it is less influenced by the punch diameter.

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## DYNAMIC TESTS PERFORMED ON FLAT FREIGHT WAGON

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### 1. Introduction

The SUSTRAIL project started in June 2011 having as main objective to provide the approach, structure, and technical content to improve the Sustainability, Competitiveness, and Availability of European railway networks thanks to an integrated approach [1]. To achieve this, a consortium of European experts was formed and considered combined improvements in both freight vehicle and track components in a holistic approach including economic assessments. Romanian Railway Authority (AFER), as a member of the consortium, had the task to perform all experimental tests necessary to validate the design and the construction of the prototype of a freight wagon.

This paper presents the experimental determinations of static and dynamic stresses in the chassis of the wagon, using the strain gauge technique. .

### 2. The dynamic test facility

All tests were done by AFER at Făurei Railway Testing Center, on big ring (Fig. 1).

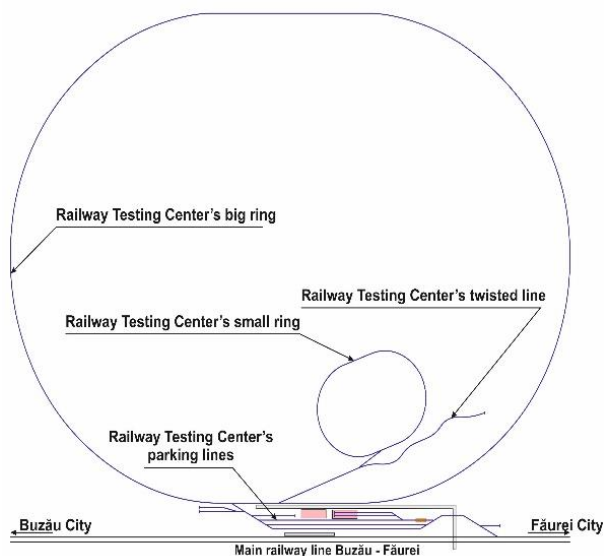


Fig. 1. AFER's Railway Testing Center.

### 3. The prototype

Based on the design proposed by the joint work of members of the consortium from universities and companies, the prototype of the freight wagon was assembled in Romania, at UVA Aiud, using parts and constituents manufactured in different factories from Europe.

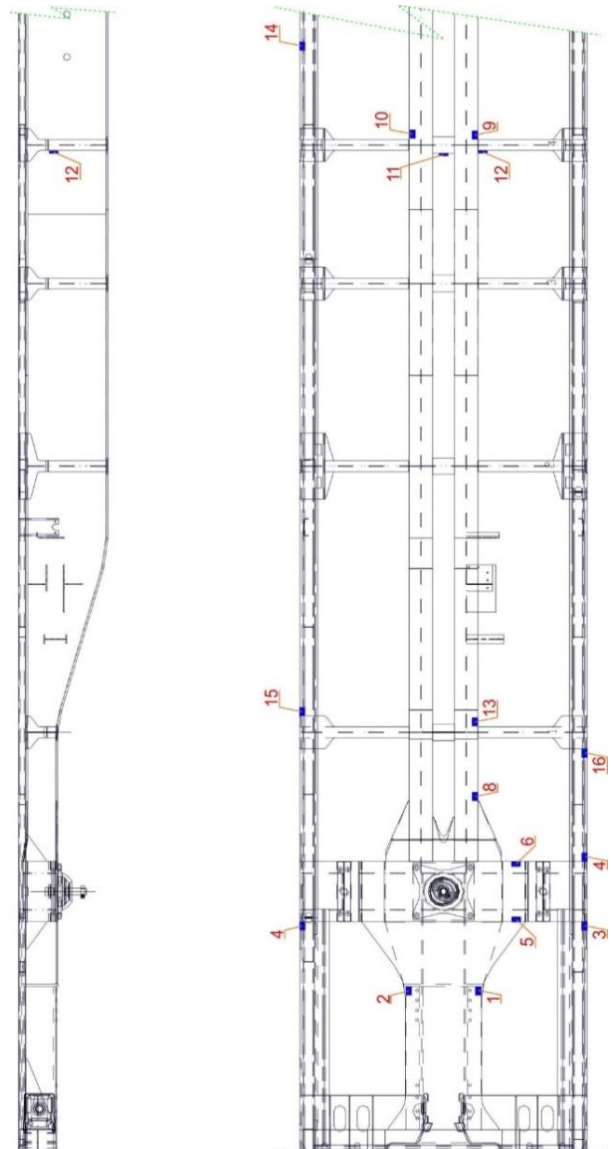


Fig. 2. Detail of the chassis and strain gauges.

16 strain gauges were glued on the chassis of the wagon [2], [4], as shown in Fig. 2. The locations of the strain gauge were chosen based on previous three dimensional finite element analyses which revealed areas with high stresses.

#### 4. The experimental set-up

Data obtained from HBM 1-LY11 10/120 strain gauges were acquired using HBM MGCplus data acquisition systems and processed with a software application specially created and customized for this project by the authors in HBM Catman Professional data acquisition software by AFER's the authors of this paper. As sensors, we used HBM 1-LY11 10/120 strain gauges.

#### 5. Tests

The tests were performed in two steps:

- static tests to find the values of the static stresses  $\sigma_{st}$ ,
- dynamic tests, in which the stress variations  $\Delta\sigma_+$  and  $\Delta\sigma_-$  were obtained during wagons's running with a speed of 140 km/h. Based on values of  $\Delta\sigma_+$  and  $\Delta\sigma_-$ , the maximum and minimum dynamic stresses were calculated as: [4]:

$$\sigma_{max} = \sigma_{st} + \Delta\sigma_+ \quad (1)$$

$$\sigma_{min} = \sigma_{st} + \Delta\sigma_- \quad (2)$$

Further, the mean stress  $\sigma_m$  and the stress amplitude  $\sigma_a$  were obtained with the equations:

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad (3)$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \quad (4)$$

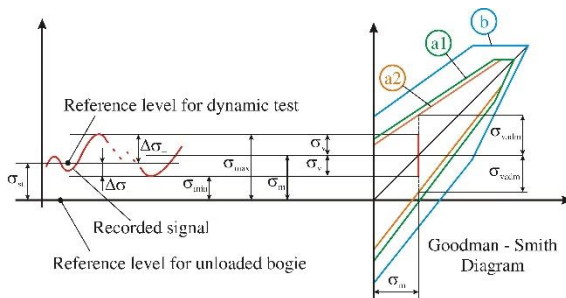


Fig. 3. The test diagram.

#### 6. Results

The obtained stress amplitudes were inserted in the Goodman – Smith diagram, taken from annex F.3 of the report ERRI B12/RP17. (see Fig. 3) The allowable condition is:

$$\sigma_v \leq \sigma_{adm} \quad (5)$$

The results for static tests (vertical load) are presented in Table 1.

Table 1. Results of the static tests

| Stress [N/mm <sup>2</sup> ] |     |     |     |     |     |      |     |
|-----------------------------|-----|-----|-----|-----|-----|------|-----|
| Strain gage                 |     |     |     |     |     |      |     |
| 1                           | 2   | 3   | 4   | 5   | 6   | 7    | 8   |
| -14                         | -6  | -47 | -75 | 25  | -26 | -174 | 113 |
| Strain gage                 |     |     |     |     |     |      |     |
| 9                           | 10  | 11  | 12  | 13  | 14  | 15   | 16  |
| 218                         | 194 | -44 | -11 | 267 | -95 | -13  | -96 |

In figure 4 is presented the recorded signal for one strain gage during test.

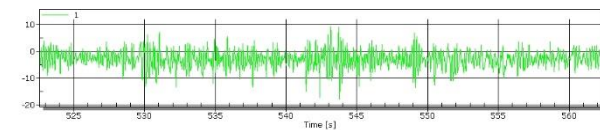


Fig. 4. Recorded signal during test, strain gage 1

#### 7. Conclusions

No residual stresses were recorded after vertical loads removal or after stopping the train at the end of dynamic test.

#### Acknowledgements

SUSTRAIL: The sustainable freight railway: Designing the freight vehicle - track system for higher delivered tonnage with improved availability at reduced cost This project has received funding from the European Union's Seventh Framework Programme for research, technological development and demonstration under grant agreement n° 265740 (<http://www.sustrail.eu/>).

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## EXPERIMENTAL STRESS ANALYSIS OF A RAILWAY CAR USING A DEDICATED TEST RIG

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### 1. Introduction

Before putting in service a railway coach, a series of tests for determination of the stress field have to be carried out according to the international standards [1], [2].

In this paper, the stress analysis of a railway car is carried out by means of strain gauges. The use of strain gauges is also required by the reference documents describing testing procedures for the load-bearing structure of railway vehicles. The objective of the presented experimental analyses is to determine the stresses in each measuring point based on the strains obtained experimentally.

### 2. Description of the test rig

Static tests must be performed for all new prototype vehicles, for vehicles which are upgraded and for the ones with changes in the strength structure.

The strain gauge measurements on the body of a wagon, described in this paper, were undertaken in the laboratory of the Romanian Railway Authority (AFER). The static test rig used for testing of railway vehicles is shown in Fig. 1.

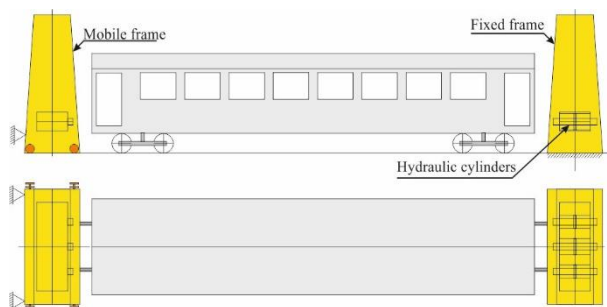


Fig. 1. Test rig - AFER.

The test rig has a fixed frame with hydraulic cylinders (for horizontal loads) and a mobile frame. The mobile frame position depends on the length of the tested body. The hydraulic cylinders are mounted in a girder which can be moved vertically. Similarly, on the mobile frame there is a girder which also can be moved on the same direction.

### 3. The studied railway car

The studied wagon body is sent without windows, doors and auxiliary facilities, in order to allow glueing of the strain gauges [1].



Fig. 3. Wagon on test rig.

The vehicle is made of S355 steel, for which the following allowable stresses should be taken into account according to [3]: 355 MPa for seamless areas and 323 MPa for welding areas.

### 4. Tests description

The standard static tests performed on the structure of the passenger car, using the strain gauges, on the test rig are:

A) Compressive force at buffer level, with 1000 kN/buffer;

B) Tensile force in coupler area, with 1500 kN force;

C) Vertical static loads for the vehicle body [3].

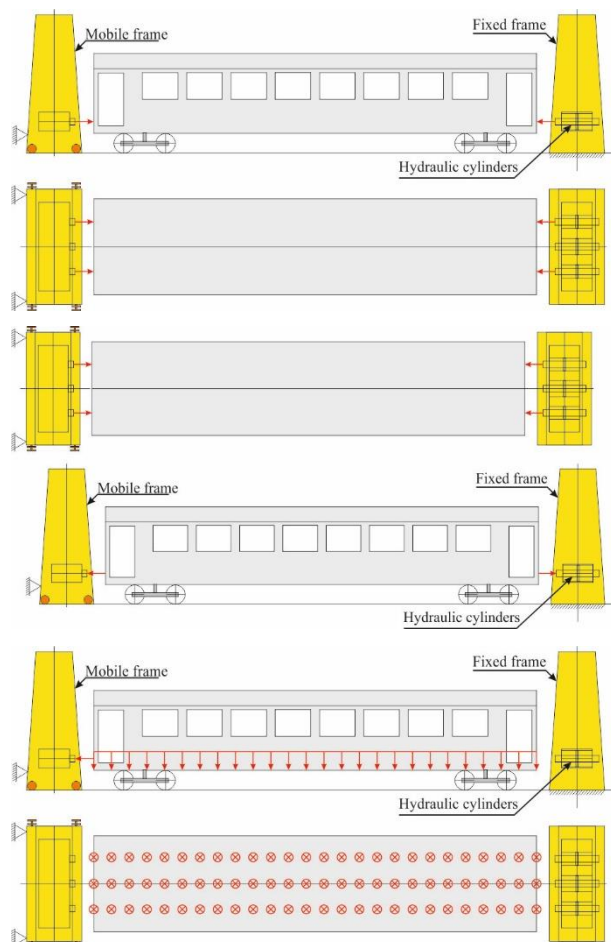


Fig. 4. Load cases.

Whereas the bodywork is not equipped for being put into use, in order to allow access to the strength structure for mounting of strain gauges, the wagon was loaded with metallic pieces with a mass of approximately 800 kg to compensate the difference between the mass of the tested structure and the mass of wagon ready to be put into service.

For test, we use data acquisition systems from HBM (Centipede 100 multipoint measuring device, computer and data acquisition software Catman). For each vehicle, a new application of data acquisition is made. The position for each strain gauge was chosen based on finite model and finite element analyses. Also some “possible hot point” are indicated in standards.

## 5. Results

The stress output of the performed measurements is listed in Table 1. For each test

case, 10 of the most relevant values were presented both at the chassis level and the wagon box level.

Table 1. Results of the static tests, stress [MPa]

| Part    | Strain gauge | A    | B   | C    |
|---------|--------------|------|-----|------|
| Chassis | 1            | -156 | 110 | -5   |
|         | 2            | 83   | -60 | -54  |
|         | 3            | 92   | -65 | -74  |
|         | 4            | -45  | 46  | -18  |
|         | 5            | -40  | 48  | -27  |
|         | 6            | -132 | 98  | -2   |
|         | 7            | 76   | -48 | -65  |
|         | 8            | 82   | -54 | -65  |
|         | 9            | -40  | 39  | -20  |
|         | 10           | -22  | 27  | -14  |
| Body    | 1            | -104 | 69  | 75   |
|         | 2            | 30   | -27 | -133 |
|         | 3            | -87  | 76  | 134  |
|         | 4            | -21  | 16  | 22   |
|         | 5            | 14   | -11 | -25  |
|         | 6            | -10  | 5   | -19  |
|         | 7            | 15   | -8  | 36   |
|         | 8            | -179 | 122 | -35  |
|         | 9            | -28  | 18  | -13  |
|         | 10           | 34   | -22 | -5   |

## 6. Conclusions

The test results show a uniform distribution of stresses along the wagon on both the chassis side and the box area.

The maximum stresses obtained experimentally do not exceed the allowable limit of the material.

Different ways of changing the wagon can be tried to reduce tare weight in the areas where the stresses are low and the constructive elements are less loaded, by reducing the thickness of certain profiles, by changing the material or even by eliminating certain elements.

This can be accomplished by performing an optimization procedure using the finite element method.

## References

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## EXPERIMENTAL STRESS ANALYSIS ON RAILWAY VEHICLE BODY

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### 1. Introduction

In order to verify aspects related to safety traffic, railway vehicles should be subjected to a series of tests according to SR EN 12663 European standards for validation before the final homologation. Railway vehicles should perform static and dynamic tests as: static and dynamic tests of carbody, brake performance tests, running dynamic behavior tests, noise tests etc. according to European regulations.

In this paper, static tests with longitudinal and vertical forces were performed on a Eaos type railway vehicle. The Eaos wagon is a open for axle freight vehicle used for goods transport like: coal, ore, timber, steel etc. Stresses were measured at selected points using the strain gauge method and compared with the allowable values according to SR EN 12663-2:2010 and vehicle material type (S355 steel).

### 2. The experimental set-up

A complex test rig, made of different steel elements and a hydraulic load control system (Fig. 1) was used for these tests. This test rig can be configured depending on the tested rail vehicle length by moving the mobile beam.

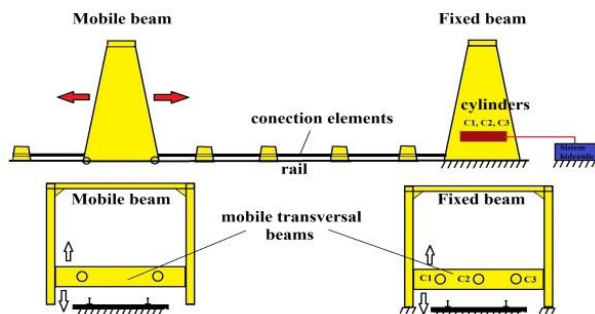


Fig. 1. Test rig

The measuring chain was composed of: unidirectional strain gauges and rosettes, cables,

connections devices, data acquisition system, laptop with Catman software.

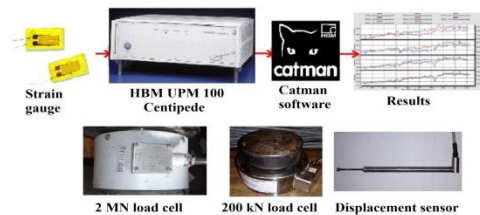


Fig. 2. Measuring devices

Forces were measured with 2 MN load cells, for the longitudinal static tests, and with 200 kN load cells for the vertical static tests.

### 3. Characteristics of the tested railway vehicle

The main characteristics of the tested wagon (Fig. 2.) are:

- Gauge: 1435 mm;
- Length of the body vehicle: 12792 mm;
- Width of the body vehicle: 2772 mm;
- Height of the body vehicle: 1890 mm;
- Body vehicle surface: 35,46 m<sup>2</sup>;
- Vehicle mass: 22 t;
- Maximum axle load: 20 t [1].



Fig. 2. Tested vehicle

### 4. Performed tests

According to standard requirements, the following tests were performed:

- Longitudinal static tests for the vehicle body:
  - compressive force at buffer and/or coupler attachment 1500 kN (CT and CA) - 1000 kN/buffer, 1500 kN for coupler attachment;
  - compressive force below buffer and/or coupling level (CT50) – 750 kN/buffer;
  - compressive force applied diagonally at buffer attachment (CDD and CDS) – 400 kN/buffer;
  - tensile force at coupler attachment (TA) –1500 kN.

- Vertical static tests for the vehicle body:
  - operating load with 568,8 kN load (SV) [2].
 An example of how loads were applied on the vehicle body is presented in Fig. 3.

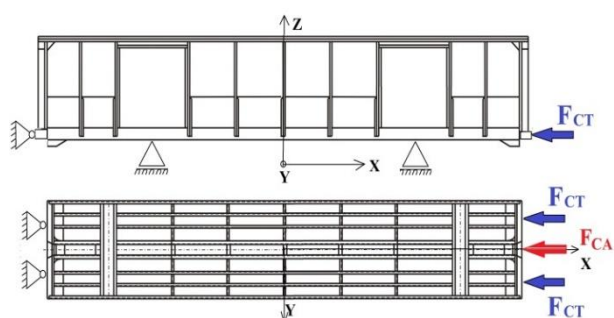


Fig. 3. Loads applied in the longitudinal static tests

A number of 55 strain gauges were mounted on the entire vehicle in order to have a picture of the stress state. Most of the transducers were mounted into the chassis area, where the main forces are acting as seen in Fig. 3.

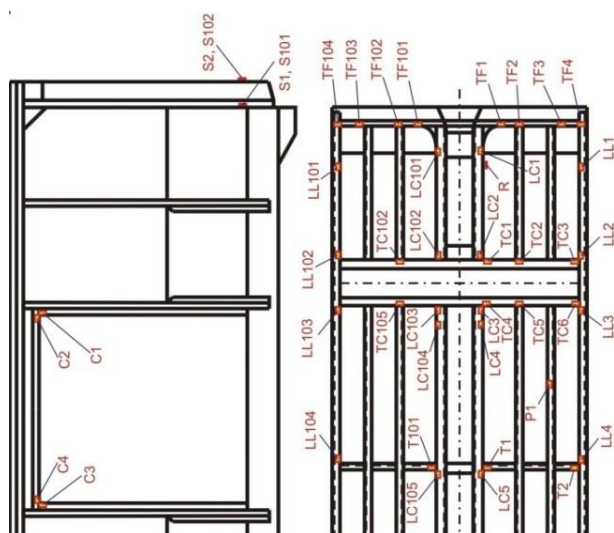


Fig. 3. Position of the most important measuring points

## 5. Test results

During tests, strains were recorded and transformed into stresses using Hooke's law.

As a result of the loads applied on the vehicle structure, through Catman software, the relative displacement values were transformed into stress values. The main results are presented in Table 1.

Table 1. Main results

| Compressive force at coupler attachment (CA) |                 |                    |                       |
|----------------------------------------------|-----------------|--------------------|-----------------------|
| No.                                          | Measuring point | Stress limit [MPa] | Measured stress [MPa] |
| 1                                            | LC103           | 322                | -45                   |
| 2                                            | LC4             | 355                | -128                  |
| 3                                            | LC104           | 355                | -115                  |
| Compressive force at buffer attachment (CT)  |                 |                    |                       |
| No.                                          | Measuring point | Stress limit [MPa] | Measured stress [MPa] |
| 1                                            | LC1             | 355                | -218                  |
| 2                                            | LC101           | 355                | -144                  |
| 3                                            | LC2             | 322                | -31                   |
| Tensile force at coupler attachment (TA)     |                 |                    |                       |
| No.                                          | Measuring point | Stress limit [MPa] | Measured stress [MPa] |
| 1                                            | LC2             | 322                | 100                   |
| 2                                            | LC102           | 322                | 108                   |
| 3                                            | LC4             | 355                | 98                    |
| Operating load (SV)                          |                 |                    |                       |
| No.                                          | Measuring point | Stress limit [MPa] | Measured stress [MPa] |
| 1                                            | LC102           | 110                | -96                   |
| 2                                            | LC4             | 277                | -90                   |
| 3                                            | C2              | 110                | -77                   |

## 6. Conclusions

Static tests with longitudinal and vertical forces were performed on an Eaos railway vehicle in accordance with the requirements of the SR EN 12663-2:2010 European standard [2]. The strain gauge method was used to determine the stresses in the considered structure vehicle. The obtained results show that the stress limits were not exceeded. Consequently, the car is safe from the point of view of the stress criterion.

## References

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## EXPERIMENTAL STUDY OF THE DYNAMIC RESPONSE OF A CABLE UNDER WIND FLOW

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### 1. Introduction

The purpose of this study is to determine the dynamic response of an inclined cable under the effects of wind blowing and extrapolating these results in order to get a more exact estimation of the behavior of real cable stays used on bridges. In order to do this, a scaled model of a cable stay was tested in a wind tunnel under different wind conditions. Numerical models of these tests were done and design parameters were corrected in order to get more precise results between the actual responses and the simulated ones.

### 2. Test model

The tests were done in the wind tunnel of the Hydraulics Faculty of the Technical University of Civil Engineering of Bucharest. It is 25.00m long with a squared section of 2.75m side.



Fig. 1. Wind tunnel and cable model

The cable was made up of a 3-wire strand, each with a diameter of 3mm yield strength of 1860MPa, encased in a 50mm sheath filled with foam, leading to a weight of 224 g/m. the cable was anchored diagonally to the corners of the tunnel thus having a length of 2.404m. Attached to the cable there were

three-dimensional accelerometers at key points in the middle and quarters of the cable.

Corresponding measurements were taken from each device in three orthogonal directions with a frequency of 1200Hz for 27 cases, considering different internal stresses in the cable and varying wind speeds between 0 and 15m/s.

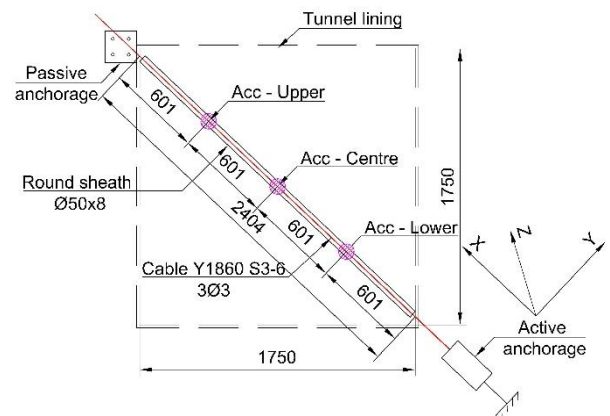


Fig. 2. Cable sketch

### 3. Numerical models

In order to simulate the dynamic response of the cable, the numerical approach consisted of 2 different finite element models. The first one was a 2D model made considering computational fluid dynamics and was used to determine the effect of the wind on the cable, such as vortex shedding and variation of lift and drag coefficients in time.

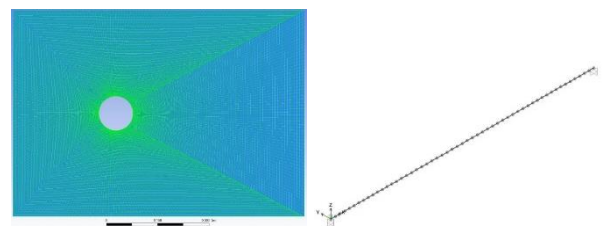


Fig. 4. Finite element models

The second model was a 3D model of the cable using 49 Timoshenko formulated beam elements with 6 degrees of freedom in each node, considering the structural mechanics domain. This was used to simulate the dynamic response of the cable under the effect of the forces predetermined from the previous model

#### 4. Results

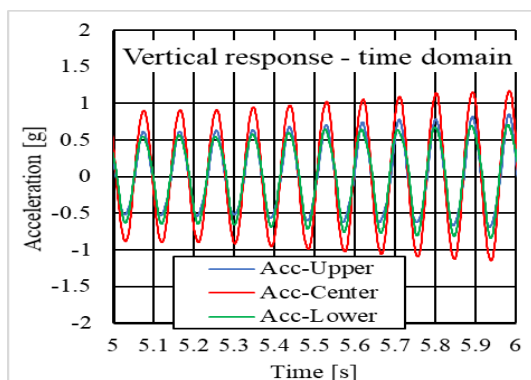


Fig. 6. Cable vertical response under wind speed of 3m/s time domain

The results were processed and analyzed separated in two directions, vertical and horizontal. Certain corrections and transformations had to be made in order to get a better understanding of the phenomena. In order to do this, two different approaches were considered, one with time-history variations of the accelerations levels in the three points in the time domain and one in the frequency domain for the same recordings.

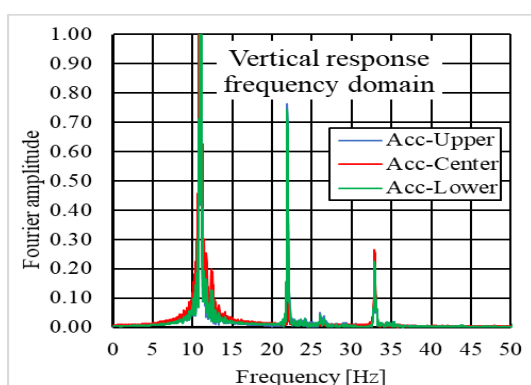


Fig. 7. Cable vertical response under wind speed of 3m/s frequency domain

#### 5. Conclusions

The vibrations due to wind blowing on the cable were significantly larger in the direction perpendicular to the air flow (vertical) as compared to the ones in the flow's direction. In most cases, the

fundamental mode of vibration was found to be dominant, but there were recordings in which the upper modes would dictate the response, especially the first three.

The highest oscillation amplitudes were recorder for relatively low wind speeds of 3 to 5 m/s, for which the shedding frequency overlapped the cable's frequency, and so the resonance phenomena were stronger.

Acceptable simulations of cable dynamic responses under wind conditions using finite element models are viable but very difficult to do, due to the large number of parameters involved. The most important ones are related to the damping model, integration method and time step size. After tuning all the design parameters, the simulated response and the recorded one were quite close, with errors around 5%.

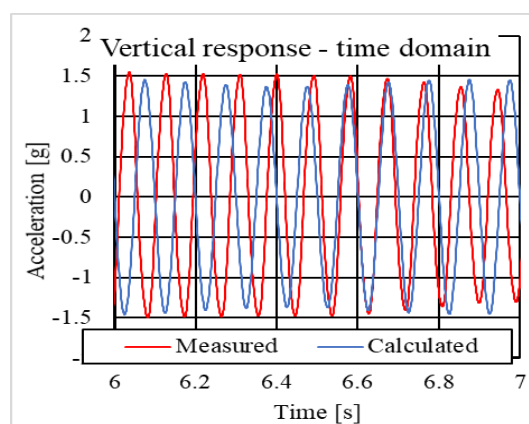


Fig. 8. Comparison between measured and estimated cable response

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# KEY FACTORS TOWARDS A HIGH QUALITY ADDITIVE MANUFACTURING (3D PRINTING) PROCESS WITH ABS MATERIAL

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## 1. Introduction

Additive Manufacturing (AM) is one of the most rampant and user-friendly manufacturing solutions which is currently redefining production technologies. Latest technological advances, created a wide range of opportunities, into various domains, from biomedical applications to the aerospace industry, both on prototypes and end-user parts with a high geometrical complexity. This diversity of applications is influenced by the possibility of using different materials and AM methods. The most common materials used for 3D printing include: metals and alloys, polymers and composites, ceramics and concrete. The type of materials used is directly linked with the 3D printing method [1]. Based on their popularity the main methods of 3D printing are: Fused deposition modelling (FDM); Laminated object manufacturing (LOM); Stereolithography (SLA); Powder bed fusion which include selective laser sintering (SLS), selective laser melting (SLM) and three-dimensional printing (3DP); Direct energy deposition (DED) which include laser solid forming (LSF), direct metal deposition (DMD) and wire + arc AM (WAAM); Inkjet printing and contour crafting used to print ceramic and concrete structures.

Because it is one of the most popular AM technologies and due to some significant advantages in terms of low equipment and material costs and low material wastage, we chose to study the main factors of influence in terms of tensile strength for the Fused Deposition Modelling (FDM) process.

## 2. Experimental methodology

This paper's aim is to identify which are the main factors that influence the tensile strength of FDM 3D printed parts and how they can be adopted in achieving improved end-user parts.

Based on a literature review [2] [3] we identified that the main parameters that influence the tensile strength of FDM parts are: layer thickness ( $L_t$ ), build orientation ( $B_o$ ), raster angle and width, infill density and pattern, extrusion temperature and printing speed - feed rate (mm/s) and flow rate (mm<sup>3</sup>/s).

The specimens used in this study are modelled in CATIA and are based on American Society for Testing and Materials ASTM D638 type IV standards for plastic tensile testing. For this experiment 3 different 3D printers were used: Makerbot Replicator 2X, UP Plus 2 and a custom made Ultimaker 3D printer, each of them using a different software: Makerbot Print 3.0, UP Studio version 2.4, and Cura version 3.3.1. Depending on the printer used, the studied processing parameters are (Fig. 1): Build orientation ( $B_o$ ): flat, on edge and vertical; Infill density ( $I_d$ ); Infill pattern; Printing temperature ( $T$ ); Print speed ( $P_s$ ); Raster angle ( $R_a$ ); Nozzle diameter ( $N_\emptyset$ ); Side wall thickness ( $V_t$ ); Layer thickness ( $L_t$ );

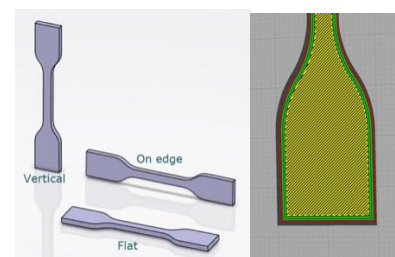


Fig. 1. Studied processing parameters

The material used for this experiment is acrylonitrile-butadiene-styrene (ABS) that comes as 1.75mm filament spools in vacuum packages from the same supplier and from the same batch. It is known that moisture absorption affects the mechanical properties of the thermoplastic polymer materials. In order to minimize this effect after the vacuum package has been unsealed all samples were printed simultaneously, in the same room which benefits from a controlled environment. The printed samples were stored at the same location and

all the tensile tests were made in one day at approximately the same environment conditions.

Because each 3D printer used a different software with different capabilities, not all the printing parameters could be modified. For example, on the UP Plus 2 printer we modified only the build orientation (flat and vertical – 17 test samples in total) keeping the other parameters constant: infill 75%, layer thickness 0.15mm, raster angle 45°, printing temperature 230°C, nozzle Ø 0.4mm, side wall thickness 3 shells, infill pattern hexagonal, print speed “fine”. When printing on the Makerbot we encountered some difficulties with the extruder and we could only print 2 sets of 5 specimens each modifying the only the build orientation from flat to side. We couldn’t print vertical samples due to the low adhesion to the printer’s platform. The other print parameters were set as follows: infill 100%, layer thickness 0.2mm, raster angle 45°, printing temperature 240°C, nozzle Ø 0.4mm, side wall thickness 3 shells, infill pattern hexagonal, print speed 90 mm/s. From all three printers the Ultimaker is the least restrictive, offering the possibility to modify almost all printing parameters. On this printer we made the most test samples, 39 in total. Each sample set consists of five specimens for a given group of parameters.

### 3. Results and Discussion

During the experiment 69 specimens are tested to failure. The results are compared to the typical ranges of mechanical properties of ABS materials fabricated by FDM technology [4]: Tensile strength: 36-71.6MPa; Young’s modulus: 99.8-2413MPa; Elongation Break ( $\epsilon$ ): 3-20%.

The sample sets properties are as described in table 1. Sample sets S1 to S3 have been printed on UP Plus 2, S4 and S5 on Makerbot Replicator 2X and all the rest on the Ultimaker.

**Table 1.** Sample set properties

| Set | B <sub>o</sub> | I <sub>d</sub><br>% | L <sub>t</sub><br>mm | V <sub>t</sub><br>mm | R <sub>a</sub><br>deg. | T<br>°C | N <sub>o</sub><br>mm | P <sub>s</sub><br>mm/s |
|-----|----------------|---------------------|----------------------|----------------------|------------------------|---------|----------------------|------------------------|
| S1  | V              | 75                  | 0.15                 | 1.2                  | 45                     | 240     | 0.4                  | Normal                 |
| S2  | S              | 75                  | 0.15                 | 1.2                  | 45                     | 240     | 0.4                  | Fine                   |
| S3  | V              | 75                  | 0.15                 | 1.2                  | 45                     | 240     | 0.4                  | Fine                   |
| S4  | S              | 100                 | 0.2                  | 1.2                  | 45                     | 240     | 0.4                  | 40                     |
| S5  | H              | 100                 | 0.2                  | 1.2                  | 45                     | 240     | 0.4                  | 40                     |
| S6  | H              | 100                 | 0.2                  | 1.5                  | 45                     | 240     | 0.4                  | 60                     |
| S7  | H              | 70                  | 0.2                  | 1.5                  | 45                     | 240     | 0.4                  | 60                     |
| S8  | H              | 50                  | 0.2                  | 1.5                  | 0-90                   | 240     | 0.8                  | 60                     |
| S9  | H              | 70                  | 0.2                  | 1.5                  | 45                     | 240     | 0.8                  | 40                     |
| S10 | H              | 70                  | 0.2                  | 1.5                  | 45                     | 240     | 0.8                  | 60                     |
| S11 | H              | 70                  | 0.2                  | 1.5                  | 0-90                   | 240     | 0.8                  | 60                     |
| S12 | S              | 75                  | 0.2                  | 1.5                  | 45                     | 240     | 0.8                  | 60                     |
| S13 | V              | 100                 | 0.15                 | 2                    | circle                 | 240     | 0.4                  | 60                     |

**Table 2.** Average test results for the sample sets

| Set | F <sub>max</sub><br>[N] | $\Delta L_{max}$<br>[mm] | $\sigma_t$<br>[MPa] | $\epsilon$<br>[%] | E<br>[MPa] |
|-----|-------------------------|--------------------------|---------------------|-------------------|------------|
| S1  | 368,5                   | 1                        | 14,8                | 0,024             | 655,6      |
| S2  | 463,9                   | 1,8                      | 19,3                | 0,04              | 483,2      |
| S3  | 400,5                   | 1                        | 16,6                | 0,02              | 693,4      |
| S4  | 467,9                   | 3,4                      | 19,6                | 0,09              | 222,4      |
| S5  | 482,4                   | 11                       | 20,1                | 0,24              | 84,4       |
| S6  | 621,4                   | 8                        | 25,9                | 0,18              | 146,7      |
| S7  | 626,7                   | 9                        | 26,1                | 0,2               | 130,5      |
| S8  | 447,8                   | 6                        | 18,6                | 0,13              | 188,1      |
| S9  | 644                     | 10,8                     | 26,8                | 0,24              | 112,1      |
| S10 | 611,2                   | 9,2                      | 25,4                | 0,2               | 122,9      |
| S11 | 630,3                   | 6,2                      | 26,2                | 0,13              | 322,8      |
| S12 | 651,7                   | 2,6                      | 27,1                | 0,06              | 434,5      |
| S13 | 382,9                   | 1                        | 15,9                | 0,02              | 797,7      |

### 4. Conclusions

Analyzing the results, we can determine that the mechanical properties are mainly influenced by the build orientation, infill pattern and density and by the printing speed. By comparing S2 with S12 we can conclude that the 3D printer nozzle diameter has a significant influence on the mechanical properties.

### Acknowledgements

This work is supported by the DiFiCIL project (contract no. 69/08.09.2016, ID P\_37\_771, web: <http://difcil.grants.ulbsibiu.ro>), co-funded by ERDF through the Competitiveness Operational Programme 2014-2020.

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## STUDY REGARDING THE GEOMETRICAL DEVIATIONS FOR 3D PRINTED FUNCTIONAL PARTS

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### 1. Introduction

Additive Manufacturing (AM) popularity is currently on an ascendant trajectory, the main cause for this course consist in an increase of 3D printing methods and available materials. Based on their popularity the main methods of 3D printing are:

- Fused deposition modelling (FDM) – a continuous filament (usually from a thermoplastic polymer) is melted, extruded and deposited into layers on top of previously created layers that subsequently form the finished part.
- Laminated object manufacturing (LOM) where successive layers of material are cut and then bounded together
- Stereolithography (SLA) uses a beam of UV light to solidify a resin layer by layer;
- Powder bed fusion which include selective laser sintering (SLS), selective laser melting (SLM) and three-dimensional printing (3DP);
- Direct energy deposition (DED) which include laser solid forming (LSF), direct metal deposition (DMD) and wire + arc AM (WAAM). The difference between DED and SLM is that DED doesn't use a powder bed, the material is melted and deposited in a similar way as FDM.
- Inkjet printing and contour crafting used to print ceramic and concrete structures from tissue engineering to concrete buildings.

In its early stages AM was used only for prototype development or demonstrators, nowadays AM tends to be used for obtaining functional parts and for small scale production. Latest development of hybrid machines which combine DED technology with CNC milling tend to push AM even further, towards mass production. This trend produces new challenges mainly related to the mechanical properties and the dimensional accuracy of the 3D printed parts [1]. Because FDM is the most common 3D printing method we chose to study the geometrical deviations resulted on representative parts. By using FDM as an AM

method high complexity parts can be achieved without the need of specialized skills. Due to its high popularity, FDM also benefits from a high support provided by the online communities. Another key factor that influenced our decision to study FDM is the possibility of creating cased functional mechanical assemblies in a single print by using dual extruders and water-soluble materials.

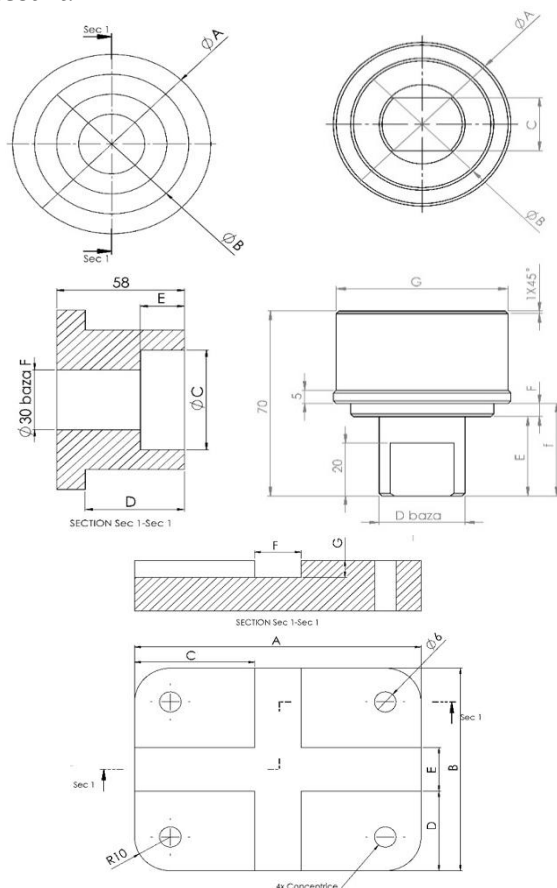
### 2. Experimental methodology

To study the geometrical deviations resulted from FDM printing method we chose three different 3D printers with different characteristics detailed as follows:

- Makerbot Replicator 2X – print software Makerbot Print 3.0, dual extruder with a nozzle diameter of 0.4 mm
- UP Plus 2 – print software UP Studio version 2.4, single extruder with a nozzle diameter of 0.4 mm
- Ultimaker PRO – print software Cura 3.3.1, dual extruder with a nozzle diameter of 0.8 mm

We tried to keep the printing parameters (Build orientation ( $B_o$ ); Infill density ( $I_d$ ); Infill pattern; Printing temperature ( $T$ ); Print speed ( $P_s$ ); Raster angle ( $R_a$ ); Side wall thickness ( $V_t$ ); Layer thickness ( $L_t$ )) as similar as the printers software permitted. The main difference consisted in the nozzle diameter. For this experiment we chose as printing material acrylonitrile-butadiene-styrene (ABS) that comes as 1.75mm filament spools. This material was chosen due to its wide range of applications but mainly because it exhibits a high temperature range between solid and liquid state which results in a high shrinkage factor during 3D printing. In order to measure geometrical deviations we have created three representative parts (P1 a flange, P2 a shaft and P3 a plate. Fig. 1) in CATIA software. Each part was printed on each 3D printer and the dimensions are measured using a FARO Gage 3D measuring arm and also a Heidenhain 3D probe mounted on a CNC. The 3D measuring arm has an accuracy of 5.8

$\mu\text{m}$  for a working distance lower than 100mm while the 3D touch probe has an accuracy  $\leq \pm 5 \mu\text{m}$ . Each part was measured 5 times on the same surface and the average of these measurements is taken into account.



**Fig. 1.** Representative parts for measuring geometrical deviations

Dimensions in [mm] from the CAD models are:

- P1 (flange): A=90, B=70, C=50, D=45, E=20, F=30
- P2 (shaft): A=62, B=50, C=20, D=30, E=30, f=30, F=5, G=60
- P3 (plate): A=80, B=60, C=33.5, D=23.5, E=13, F=13, G=5

### 3. Results and Discussion

The average dimensions and deviations (%) for each sample are listed in Tables 1-3.

### 4. Conclusions

As a result of the experiments carried out, an increase of the reduction coefficient was determined at the increase of the wall thickness of the part. This phenomenon is explicable because the cooling time of a thicker wall is higher, during which the internal stress leading to the deformation takes effect.

**Table 1.** Average dimensions and deviations for P1

| Dim. | UP Plus 2 |       | Makerbot |       | Ultimaker |       |
|------|-----------|-------|----------|-------|-----------|-------|
|      | Dim.      | Dev.  | Dim.     | Dev.  | Dim.      | Dev.  |
| A    | 90.05     | 0.06  | 88.79    | -1.34 | 89.34     | -0.73 |
| B    | 70.05     | 0.08  | 69.35    | -0.92 | 69.61     | -0.55 |
| C    | 49.85     | -0.30 | 49.90    | -0.18 | 49.78     | -0.44 |
| D    | 45.05     | 0.11  | 45.01    | 0.03  | 45.01     | 0.03  |
| E    | 20.21     | 1.05  | 20.20    | 1.02  | 19.99     | -0.04 |
| F    | 29.71     | -0.95 | 29.90    | -0.32 | 29.68     | -1.07 |

**Table 2.** Average dimensions and deviations for P2

| Dim. | UP Plus 2 |       | Makerbot |       |
|------|-----------|-------|----------|-------|
|      | Dim.      | Dev.  | Dim.     | Dev.  |
| A    | 61.460    | -0.87 | 61.488   | -0.83 |
| B    | 50.015    | 0.03  | 49.330   | -1.34 |
| C    | 20.141    | 0.70  | 19.419   | -2.91 |
| D    | 30.016    | 0.05  | 29.490   | -1.70 |
| E    | 30.111    | 0.37  | 29.067   | -3.11 |
| F    | 35.118    | 0.34  | 34.844   | -0.45 |
| G    | 4.996     | -0.08 | 4.954    | -0.92 |

**Table 3.** Average dimensions and deviations for P3

| Dim. | UP Plus 2 |       | Makerbot |       | Ultimaker |       |
|------|-----------|-------|----------|-------|-----------|-------|
|      | Dim.      | Dev.  | Dim.     | Dev.  | Dim.      | Dev.  |
| A    | 80,03     | 0,04  | 79,07    | -1,16 | 79,70     | -0,38 |
| B    | 60,11     | 0,19  | 59,05    | -1,59 | 59,65     | -0,58 |
| C    | 33,50     | 0,01  | 33,21    | -0,86 | 33,59     | 0,27  |
| D    | 23,31     | -0,82 | 23,04    | -1,97 | 23,58     | 0,35  |
| E    | 13,07     | 0,57  | 13,20    | 1,52  | 12,85     | -1,17 |
| F    | 13,02     | 0,14  | 12,82    | -1,37 | 12,91     | -0,73 |
| G    | 5,02      | 0,34  | 5,05     | 0,96  | 4,98      | -0,46 |

### Acknowledgements

This work is supported by the DiFiCIL project (contract no. 69/08.09.2016, ID P\_37\_771, web: <http://dificil.grants.ulbsibiu.ro>), co-funded by ERDF through the Competitiveness Operational Programme 2014-2020.

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# EXPERIMENTAL STUDIES OF A BRIDGE MADE FROM REINFORCED CONCRETE BEAMS - BRIDGE OF TRANSILVANIA HIGHWAY

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## 1. Introduction

The Virtual Instrument (VI) developed on the basis of LabView® ver. 8.5 from National Instruments was used for the modal analysis in site of Transilvania highway bridge and for the identification of modal damping ratio of the system with elastomeric bearing system.

The VI can be defined as a complex software and hardware interface that is implemented in PCs, so the user can interact with traditional analysis tools. The VI panel is displayed on the computer monitor, which can look just like traditional instrument panel. VI panel has a background program which simulates the controls of the real instruments, allowing the working routines of acquisition, data analysis, graphic presentation and storage of data or of the results files.



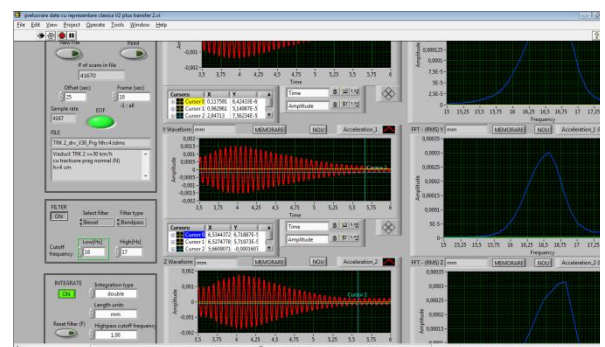
Fig. 1. Bridge of Transilvania Highway  
km 29+602,75 – km 29+801,25

VI construction is done in accordance with the user's „specifications”. In order to do that, the programme uses standard hardware architectures, proper drivers, specific software packages and special libraries. The VI uses digital tools like traditional software modules from a large package, difference being that while traditional instruments

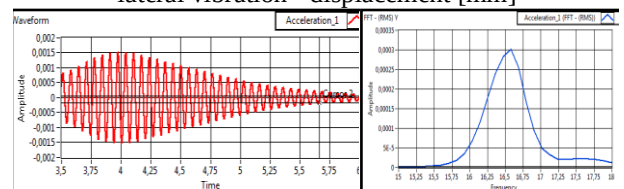
include the software in its ROM memory, the virtual instrument uses the computer HDD to do it.

## 2. Testing equipments

### MODAL ANALYSIS – VIRTUAL INSTRUMENTATION



lateral vibration - displacement [mm]



time domain [s]

frequency domain  
[Hz]

### MODAL ANALYSIS (DECOUPLED MOVINGS)

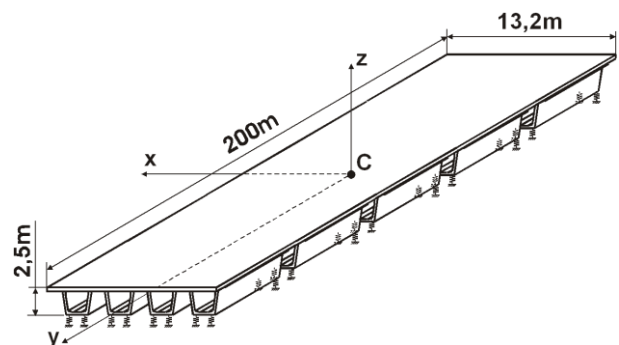


Fig. 2 The sketch of the Viaduct on Transilvania highway (axonometric view).

### 3. Results

Modal analysis for the Viaduct with decoupled movings (calculated):

| Subsystem         | Natural Frequencies [Hz]     |               | Eigenvalues [rad/m]         |                  |
|-------------------|------------------------------|---------------|-----------------------------|------------------|
| (X, $\varphi_y$ ) | $f_1 = 1,13$                 | $f_3 = 15,57$ | $\mu_1 = 0,5 \cdot 10^{-4}$ | $\mu_2 = -128,8$ |
| (Y, $\varphi_x$ ) | $f_3 = 1,13$                 | $f_4 = 16,79$ | $\mu_3 = 0,2 \cdot 10^{-6}$ | $\mu_4 = 149,9$  |
| (Z)               | $f_5 = f_z = 16,30$          |               | —                           |                  |
| ( $\varphi_z$ )   | $f_5 = f_{\varphi z} = 1,17$ |               | —                           |                  |

### 4. Acknowledgements

Experimental researches, virtual instrumentation and data analysis were made in the Vibration and Acoustic Laboratory of the Research Institute for Construction Equipment and Technology - ICECON S.A. Bucharest and the Research Center of Machines, Mechanic and Technological Equipments – MECMET from Engineering and Agriculture Faculty of Braila, “Dunarea de Jos” University of Galati.

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Jos” University of Galati, Fascicle XIV Mechanical Engineering Vol. 2 Issue XVI, 2010



## EVALUATION OF ELASTOMERIC STRUCTURAL BEARINGS PERFORMANCE BASED ON NATURAL SCALE TESTS

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### 1. Introduction

The paper presents the necessary clarifications regarding the testing and validation methods of the essential performance characteristics for the elastomeric bearings, which are necessary for the homogeneous and consistent application of the criteria and requirements specified by the technical regulations in force, and for the assessment of the use capacity for a supporting solution of a certain building structures.

According to the standard EN 1337-3 there are defined two types of tests: those for the determination of the characteristic performances (shear modulus - Annex F, compression secant modulus and compression stiffness - Annex H, rotation angle - Annex J and restoring moment - Annex K) and those for the durability in service verification of (shear resistance of bonding between the elastomer and the steel reinforcements - Annex G, compression secant modulus after dynamic fatigue - Annex I, ozone resistance - Annex L).

### 2. Objectives

Since the performance level associated with the essential characteristics of elastomeric bearings is determined based on the functional requirements and the use conditions specific for a particular building or construction work, the standard EN 1337-3 impose, in a conventional manner, that the manufacturer declared performances to be accordingly to the ultimate limit state (SLU), as defined in chapter 5.3.3. Therefore, in order to evaluate the constancy of performance for an elastomeric device manufactured by a particular manufacturer, or to compare similar products placed on the market by different manufacturers, it is essential to correctly determine the test conditions for each type of product size.

In the paper are analyzed the experimental results obtained in the testing of different types of elastomeric bearings, compared with the calculated performance using the design elements specified in EN 1337-3. The purpose of this analysis is to create a database, necessary to establish a mathematical model, based on which the performances of the similar products to be predictable evaluated, with an appropriate precision.

In order to ensure the functional requirements, the safety and durability in use, for the structures supported by the elastomeric bearings, additional verifications are required regarding the elastomeric bearings behavior in various stress conditions that may arise during the service life. These additional verifications may be performed using combined methods consisting in predictive evaluation calculations and experimental tests.

### 3. Testing equipments

The testing equipment consists in a control system for the command and monitoring of test conditions (force, displacement, time) and a hydraulic compressing machine with two independent lines of action.



**Fig. 1.** Testing Equipment – ICECON Laboratory

## 4. Results

The experimental results obtained in compliance with the ISO / IEC 17025 standard requirements are presented in test reports that contain all the information necessary to assess the products performances. These data are analyzed using statistical methods for each constructive solution (type-size) defined according to the product standard.

Finally, the obtained results are compared with the design performances calculated according to the requirements of EN 1337-3 standard, chapter 5.3.3, in order to establish the correlation coefficients which allow a more accurate mathematical modeling of the elastomeric bearings behavior.

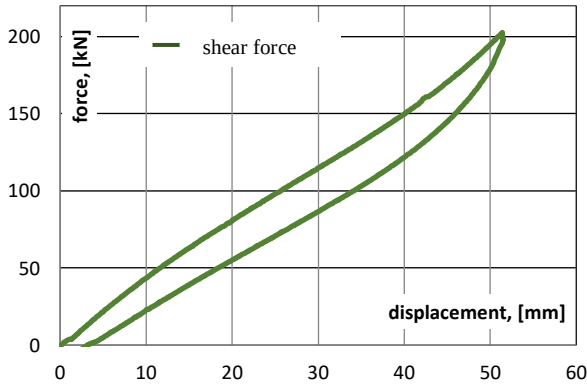


Fig. 2. Shear test results

$$\tau = \frac{F_x}{A}; \quad \varepsilon_x = \frac{v_x}{T_q}; \quad G_g = \frac{\tau_{x2} - \tau_{x1}}{\varepsilon_{x2} - \varepsilon_{x1}}; \quad (1)$$

$$R_{xy} = \frac{A \cdot G_g \cdot v_{xy}}{T_e}; \quad \varepsilon_q = \frac{R_{xy}}{A \cdot G_g}$$

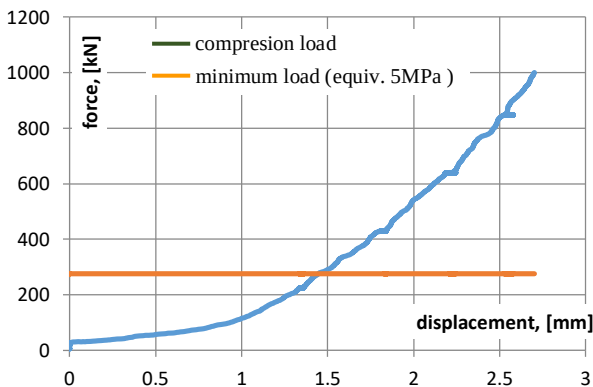


Fig. 3. Compression test results

$$\varepsilon_c = \frac{v_z}{T_q}; \quad E_{cs} = \frac{\sigma_{c2} - \sigma_{c1}}{\varepsilon_{c2} - \varepsilon_{c1}}; \quad C_c = \frac{F_{z2} - F_{z1}}{v_{z2} - v_{z1}}; \quad (2)$$

$$k_z = \frac{E_c \cdot A'}{T_q}; \quad E_c = 1/(1/E'_c + 1/E_b)$$

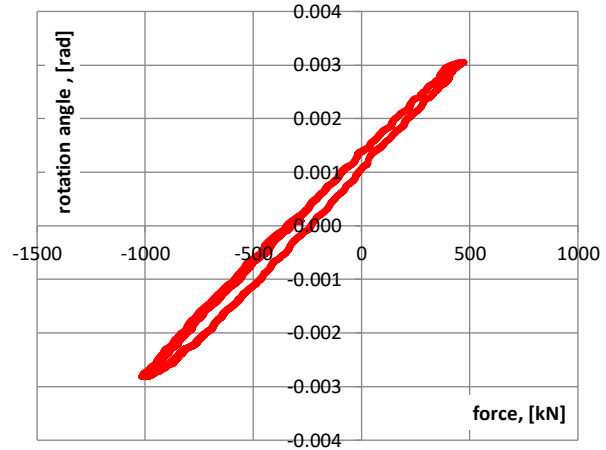


Fig. 4. Rotation / bending test results

$$M_e = \left( \frac{F_{z1} - F_{z2}}{4} \cdot L \right); \quad M = G_g \frac{\alpha \cdot a'^5 \cdot b'}{n \cdot t_i^3 \cdot K_s} \quad (3)$$

## 5. Conclusions

The analysis of the available data led to the following conclusions:

- the conventional shear modulus  $G_g$ , depends on the magnitude of the displacement  $V_{xy}$  and the maximum force  $R_{xy}$ , determined in correlation with the intended use domain;
- the compression performance (maximum displacement due to compression  $V_c$ , compression secant modulus  $E_{cs}$ , compression stiffness  $C_c$ ) are directly influenced by the magnitude of the vertical design force  $F_z$ ;
- the compressive stiffness  $K_v$  value corresponds to the experimentally determined compressive stiffness value  $C_c$ , if the tests were performed in accordance with the load conditions considered for determining the bulk modulus of elasticity;
- there are no differences between the performance of the elastomeric bearings type B and type C, because these characteristics depend only on the size and number of elastomeric layers in the product;
- the validation of the performances for the new elastomeric bearings introduced into the fabrication requires that a number of tests be carried out in order to assess the impact of the technological production processes implemented by the manufacturer.

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## ASSESSMENT OF SOIL STRENGTH AND DEFORMABILITY PARAMETERS IN LABORATORY CONDITIONS

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### 1. Introduction

The stability of buildings and infrastructures depends on compressibility and strength of soil foundation. In this regard, it is necessary to know the geotechnical soil parameters in order to determine the most reliable and sustainable measures that need to be considered for ensuring the durability and safety of in engineering works. The necessity of soil compaction process in the form of controlled fillers has occurred due to the possibility of obtaining soil layers with improved deformability, strength and bearing capacity characteristics. In the context of actual strategies worldwide adopted concerning the sustainable development practices, the technical solutions and innovative products used in various experimental applications have revealed several methodologies to optimize the functional and environmental performances linked to construction works.

The aim of the present paper is to implement eco-efficient methodologies using eco-friendly alternative materials for further application in soil consolidation, road works and industrial platforms, improvement works for railways infrastructure.

Analysis and processing of data gathered from laboratory testing carried out on natural and stabilized samples with different materials, for soil strength determination by uniaxial compression test have been performed. Soil samples were prepared in laboratory conditions by compacting the soil at optimum moisture content corresponding to each soil type, maintained on normal conditions of temperature and tested after exposure at different ages.

### 2. Materials and experimental testing

The experimental applications consisted of stabilization using laboratory treatment with different percentages of stabilizing agents by

considering soil categories classified as unstable structures. The configuration and modeling of mixtures types based on composite products have allowed the assessment of performance characteristics gained after stabilization in terms of strength and bearing capacity by performing compaction tests and compression strengths, according to mixture percentages and curing time.

Laboratory experimental testing consists in:

- setting the materials to be used in admixtures and percentages of stabilizing agents;
- preparation of soil-binder mixtures and specimens for testing;
- compaction, sampling and conditioning of samples;
- compression strength tests at different time exposure (7, 14 and 28 days) maintained in normal temperature conditions.

Two types of soils were selected for the study: sandy clay with medium plasticity; clayey sand with low plasticity. Processing and analysis of strength values corresponding to natural and stabilized samples were carried out, as illustrated in Fig. 1 and Fig. 2.

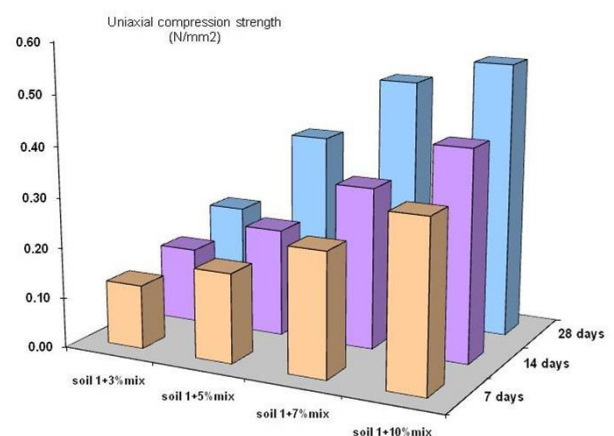


Fig. 1. Variability of strength data for soil type 1 (sandy clay)

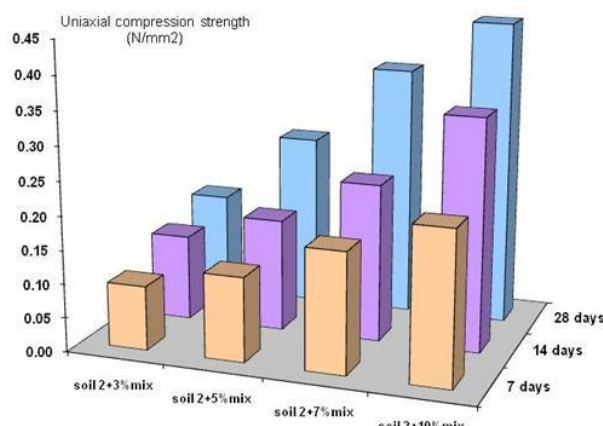


Fig. 2. Variability of strength data for soil type 2 (clayey sand)

### 3. Results and discussions

The research studies have highlighted the improvement of compaction characteristics, significant increases in workability and bearing capacity of soil foundation, and also superior resistances at different curing time, as a result of small-scale stabilization procedure using alternative materials with ecological potential.

Laboratory results based on laboratory tests on two soil types stabilized with ecological materials led to the following aspects:

- significant improvement of soil compaction characteristics and compression strength for analyzed soil types (sandy clay and clayey sand) in admixtures with stabilizing agent.

### 4. Conclusions

Performances of road layers obtained by soil stabilization process using different composite mixtures by improving of strength and deformation soil characteristics:

- achievement of high strength values corresponding to stabilized soil exposed to normal conditions of temperature and tested after exposure at different ages by comparing the values obtained for natural soil;
- ability of stabilized agent to fill the pores with a mass that gelling and solidifies over time, ensuring a high stability of road structures;
- ensuring a good workability and increase of road layers durability in time;
- flexible and advanced alternative compared to traditional solutions and technologies used for stabilization;

- reducing the works execution time;
- special maintenance operations during road system lifetime are not required.

### Acknowledgements

The paper is a part of results from experimental tests carried out within the project “Laboratory tests performed on stabilized soils from experimental polygon located in Moldova Republic, Criuleni district”.

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# SPECIALISED TESTS FOR THE ASSESSMENT AND CLASSIFICATION OF SAFES FOR THE VALUES PROTECTION, FROM THE SECURITY IN USE POINT OF VIEW

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## 1. Introduction

In order to assess and classify the performance of resistance to burglary for safes there were developed dedicated methods of tests. The peculiarity of these methods is that they reproduce, as much as it is possible and in laboratory conditions, the burglary attacks that effectively occur during their use.

From the point of view of the protection against burglary attacks, the safes can be divided into two main categories depending on the adopted constructive solutions:

- with protection against burglary depending only upon the materials and construction of its primary manufacture and not upon materials added or attached during installation;
- with protection against burglary depending partly upon the materials incorporated into it, or attached to it, during installation (wall-mounted safes or embedded in construction parts resistant to certain types of attack, specially designed for that purpose, namely wall-mounted A.T.M.s, or those installed in kiosk constructions).

In this paper there are tackled the free-standing safes, built-in safes (floor and wall) and ATM safes as storage units which protects their contents against burglary.

The tests conducted on such products are:

- the determination of the resistance against attacks, by pulling or pushing, aiming to remove the safe from the construction element to which it is anchored (Figure 1, 2);
- manual attack with burglary tools for partial access (opening through the burglary protection which enables access to only some of the safe protected content) (Figure 3, 4) ;

- manual attack with burglary tools for the complete access (opening through the burglary protection which enables access to the entire safe protected content), or the removal of the entire safe from the material in which it is embedded (Figure 5, 6).

In order to ensure a degree of test repeatability there are used tensile test equipment and burglary attack tools with similar construction and performances.

When subjected to a tensile load of 50 kN (respectively 100 kN for the upper grades), applied in the horizontal plane (case of ATM safes) or vertically (in the case of free standing safes), the construction of the safe and the anchoring assembly have to withstand the test.



Fig. 1.



Fig. 2.

For the manual attacks with burglary tools there are registered the operating times for each tool that is used during the attack and the resistance value  $V_R$  is calculated using the following formula:

$$V_R = (\sum t \times c) + \sum BV \quad (1)$$

where:

$\sum t$  is the sum of all operating times in minutes;

$c$  is the highest tool coefficient of the used attack tools (the coefficient,  $c$ , is established by the testing procedure for each family of tools);

$\sum BV$  is the sum of the basic values for all attack tools used (the basic value,  $BV$ , is established by each type of tool).

$\sum BV$  is the sum of the basic values for all attack tools used (the basic value  $BV$  is established by each type of tool).



Fig. 3.



Fig. 4.

Depending on the values obtained from the three tests, it is possible to achieve a scale in strength classes safes, quantifying perform safety use. Based on the resistance values obtained in the manual attacks and the resistance of the anchoring system,

the safes are classified in resistance grades, which represent their safety in use performance.



Fig. 5.



Fig. 6.

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# HARDWARE IN THE LOOP SIMULATION FOR HOMING MISSILES

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## 1. Introduction

Missile development is a very lengthy and costly process. To minimize expenses it is important to test all characteristics of guidance navigation and control (GNC) system in the laboratory prior to flight testing. Laboratory testing usually consists of calibration and testing of each subsystem separately as well as final testing of the system by means of Hardware-in-the-loop (HWIL) simulation [1].

HWIL simulation is a technique that is used in the development and testing of real-time embedded systems [2]. HWIL simulation provides effective information of subsystem interactions. The complexity of the system under control is simulated by adding a mathematical model of subsystems or by using subsystems themselves. Missile flight is usually simulated by its mathematical representation which runs on a computer with real time operating system (RTOS) that interacts with GNC system [3].

HWIL testing of GNC system for missiles with homing guidance usually requires five axis motion simulator. Three inner axes are used for missile attitude simulation and two outer axes simulate relative position of the target regarding the missile. Here will be presented how to perform HWIL testing with three axes motion simulator in the case of image processing homing head (HH), either Television (TV) or Infra-Red (IR) homing head.

## 2. HWIL testing on the five axes motion simulator

HWIL test setup of GNC system with homing head usually consists of: five axes motion simulator, guidance and navigation block mounted on the motion simulator, target source/simulator mounted on the external axes of the motion simulator, actuators mounted on test benches and real time computer for six degrees of freedom (6DOF) simulation of the missile and the target movement.

Guidance and navigation block consists of on-board (mission) computer (OBC), inertial measurement unit (IMU) and the homing head. Additional PC for collecting all data is usually required for later analyses of system response. Test setup is shown in Fig. 1.

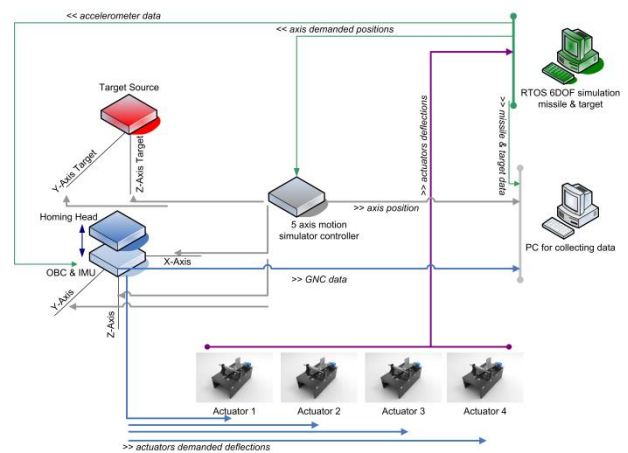


Fig. 1. HWIL setup on five axes motion simulator

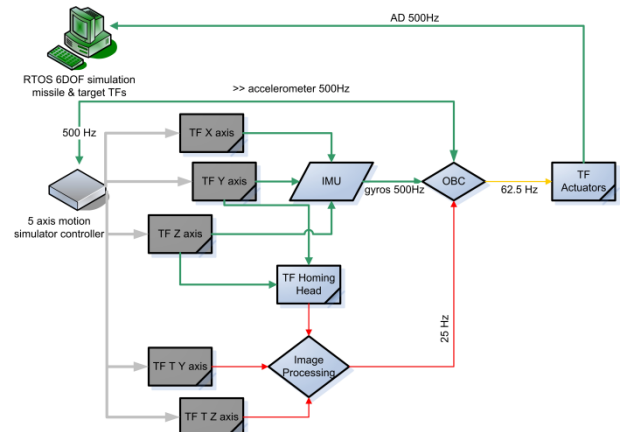


Fig. 2. HWIL time dependencies on five axes motion simulator

HWIL time dependencies for this case are shown in Fig. 2. Gray blocks represent transfer functions (TF) of five simulator axes. It is advised that the dynamic response of the simulator is at least twice higher than the dynamic response of the simulated

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196

## SECTION V: NUMERICAL METHODS



# SEISMIC CALCULATION FOR GENERATOR PROTECTION PANEL

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## 1. Introduction

Nowadays there is much more people living in earthquake zones then two or three decade before. That way the new buildings and equipment in that objects need to be checked before it will be made[1,2].

The study shown in that article has covered the following issue as: Seismic calculation for generator protection panel according to seismic zone IV acc. with National Structural Code of the Philippines (NSCP) 2010. The main aim of the project is to perform FEA check calculations, modify weak areas (if any) of the structure and provide FEA confirming correctness of design.

## 2. Geometrical and FEM model

On the base of existing cubicle shown in the figure 1, the geometrical and FEM model were prepared. Based on 3-D models of separate elements of 3D models cubicle and base, shell special models were created. Shell models faithfully represent the real object, which strongly effect on the results. Secondary structure such as bolt and nuts, mechanical or electrical equipment supports were created with the use of beam elements, electrical elements were created as concentrated mass points.

The geometrical modelling was done with the use of professional licensed CAD/CAE systems NX 10. In the figures 2 there is presented views of the cubicle and base structure with marked thickness of separate elements, enable the geometry verification of the geometrical model.



**Fig. 1.** Existing generator protection panel.

Thickness - Element-Nodal, Unaveraged, Scalar  
Min : 0.100, Max : 8.500, Units = mm

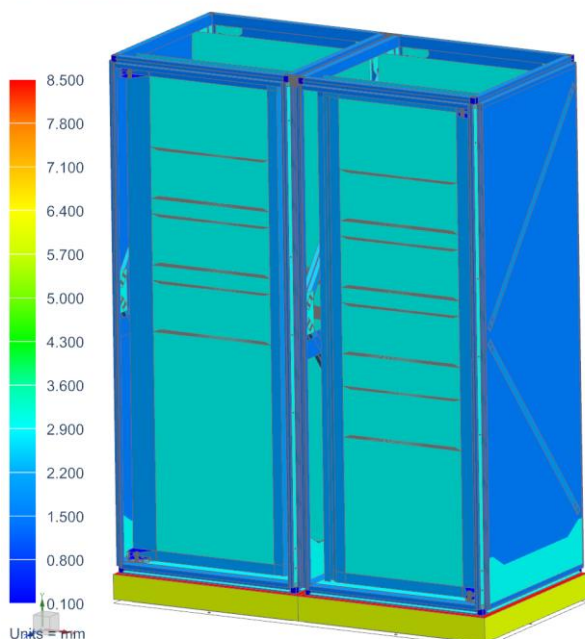


Fig. 2. FEM model.

### 3. Loads

Loads are evaluated in consideration with the standards according to seismic zone IV according with National Structural Code of the Philippines (NSCP) 2010. The acceleration loads are shown in table 1[3].

Table 1. Acceleration loads

| Case no. | Direction                                  | Acceleration                          |
|----------|--------------------------------------------|---------------------------------------|
| 1        | Horizontal X<br>Direction<br>North - South | 0,41 g ( = 4,02<br>m/s <sup>2</sup> ) |
| 2        | Horizontal Y<br>Direction<br>East - West   | 0,36 g ( = 3,47<br>m/s <sup>2</sup> ) |
| 3        | Vertical Z<br>Direction<br>Up - Down       | 0,20 g ( = 1,91<br>m/s <sup>2</sup> ) |

### 4. Results

Calculations were done in accordance with the standards. The check covers strength analysis, real eigenvalue analysis and buckling resistance of the cubicle with base. Calculations were done for the eight loads combinations. In the figure 3 there are shown the Displacement plot in mm of load set III. In the table 2 the results of FEA are presented.

Subcase - Static Loads 1 East, Static Stop 1  
Displacement - Nodal, Magnitude  
Min : 0.000, Max : 0.927, Units = mm  
Deformation : Displacement - Nodal Magnitude

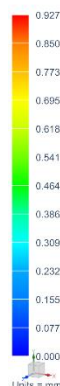


Fig. 3. Displacement plot in mm – set of 2 cubicles – load set III.

Table 2. Results of FEA

| Set of loads | Stress [MPa] | Displacement [mm] |
|--------------|--------------|-------------------|
| I            | 95,97        | 1,036             |
| II           | 115,60       | 1,181             |
| III          | 138,49       | 0,927             |
| IV           | 93,60        | 0,958             |
| V            | 158,12       | 1,041             |
| VI           | 160,40       | 1,062             |
| VII          | 169,25       | 1,264             |
| VIII         | 161,78       | 1,401             |

### 5. Conclusions

Based on preformed FEA structural calculations check, it can be confirmed that the generator protection panel consisting of 2 cubicles with the additional parts anchored with the indicated anchor can be used in seismic zone IV acc. with National Structural Code of the Philippines (NSCP) 2010.

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# ANISOTROPY OF PERIODIC MICROSTRUCTURE IN MODELS OF UNIDIRECTIONAL COMPOSITE MATERIALS

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## 1. Introduction

The layers of unidirectional (UD) continuous fiber-reinforced composites or the individual strands of textile composites are usually considered to be transversely isotropic materials. The approach is based on the assumption that the fibers are distributed regularly in perfect hexagonal pattern in the plane transverse to fiber direction and that all fibers have the same cylindrical shape and diameter. However, both the distribution and fiber shape in real material samples are always irregular (see Fig. 1), which can lead to significant difference in the predicted elasticity properties when using either analytical approaches, such as simple rule of mixtures or homogenization techniques, or numerical approaches using periodic unit cell models. The only parameter that plays role in this analysis in case of the regular distribution is the fiber volume content (or ratio)  $v_f$ .

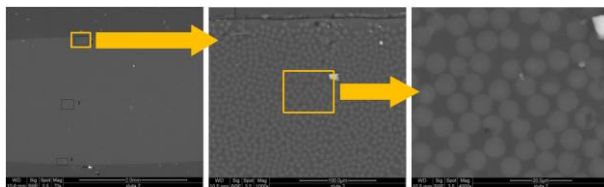


Fig. 10. Microstructure of UD composite from SEM.

In order to quantitatively describe the irregular microstructure, a new parameter  $\Upsilon$  designated as the degree of irregularity (DOI) was proposed in [1] as

$$\Upsilon(V_f, R) = 100 \times \frac{A_{\text{PUC}}^{\text{elem}}}{A_{\text{PUC}}^{\text{reg}}} \sum_{i=1}^n \left| \frac{A_{\text{AOI}}^i - A_{\text{AOI}}^{\text{reg}}}{A_{\text{AOI}}^{\text{reg}}} \right| \quad (1)$$

The value of this parameter is influenced by variation of fiber diameters, misplaced fiber positions, missing or additional fibers, and it is rotation independent. It is defined as sum over all fibers in RVE of squared differences between areas of intersection (AOI) of group of fibers between irregular and regular distributions normalized by cell dimensions (size). The mentioned areas are schematically shown in Fig. 2.

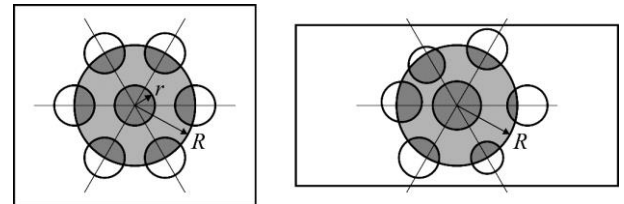


Fig. 11. Scheme of DOI calculation.

## 2. FEA model of PUC with PBC

Finite element models of the repeated periodic unit cell (PUC) can be used to study the effective or homogenized properties of the composite. An example of the mesh of the smallest PUC having 2 fibers and a larger PUC with 56 fibers in total are shown in Fig. 3.

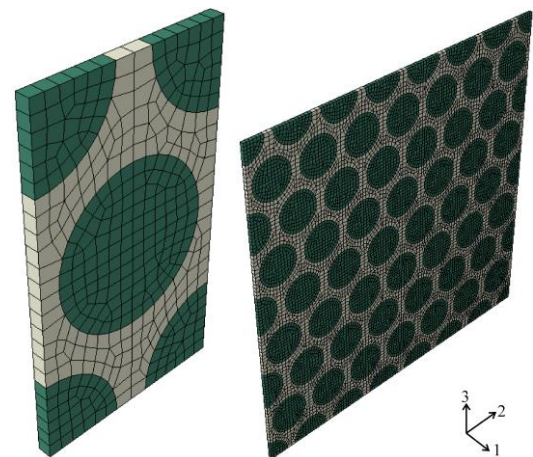


Fig. 12. Example of 2 different PUC meshes ( $v_f = 60\%$ ).

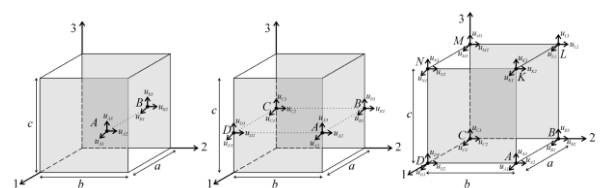


Fig. 13. Scheme of applied PBC.

The FEA model has to be constrained using appropriate periodic boundary condition (PBC). The equations applied to nodes on opposing sides,

edges or corners (see Fig. 4) ensure that the structure remains periodic even after loading [2].

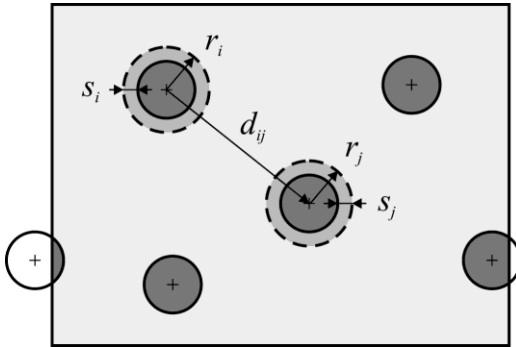


Fig. 14. Scheme of generation of random PUCs.

### 3. Random PUC generator

To study the effect of irregularity a large number of random distributions ( $0 < \Upsilon < 70$ ) all having the same fiber volume ratio ( $v_f = 60\%$ ) was generated. The geometries were created using pseudo-physics algorithm resembling the motion of magnetic particles with random attraction or repulsion fields and collision checking. Depending on each particle's parameters (Fig. 5) it is possible to generate both the regular (Fig. 6) and random distributions (Fig. 7).

### 4. Anisotropy

The resulting anisotropy of the microstructure can be expressed using the effective elastic constants such as the Young's and shear moduli, Poisson's constants, Chentsov coefficients, coefficients of interaction [3]. The values depend on the chosen coordinate system (orientation).

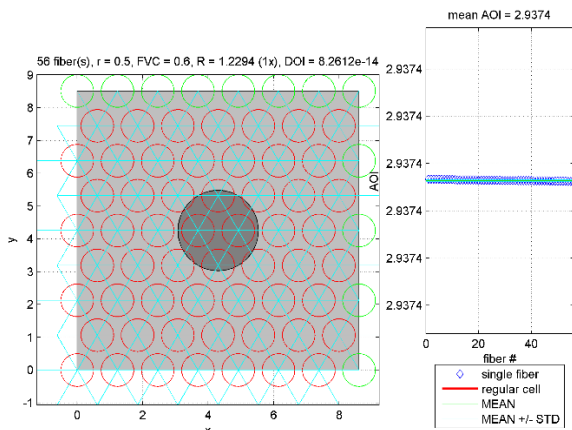


Fig. 15. Generated PUC with regular distribution.

An example of values of Young's modulus in the cross-section plane is shown for the generated random distributions mentioned above in Fig. 8. It

is obvious that the cross-section plane is no longer a plane of isotropy – it is so only in the case of regular distribution, i.e. for  $\Upsilon = 0$ , when the Young's modulus is constant (orientation independent).

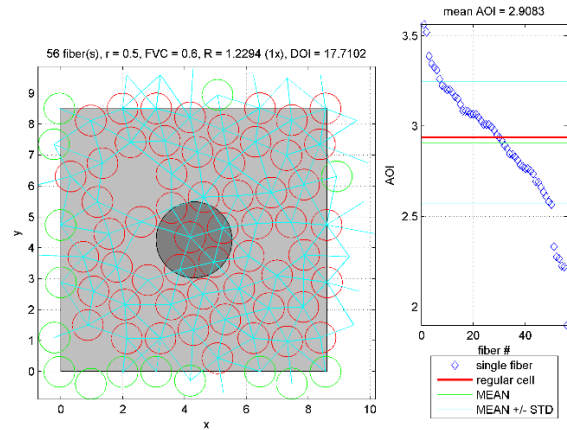


Fig. 16. Generated PUC with random distribution.

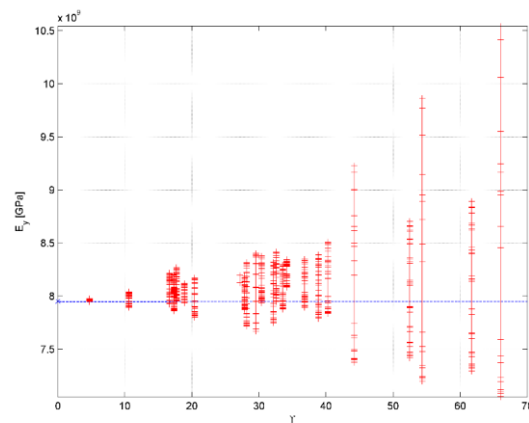


Fig. 17. Effective values of Young's modulus for various random fiber distributions (values of DOI) and orientation in the cross-section plane.

### Acknowledgements

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## ESTIMATIONS OF THE SHIP HULL OSCILLATIONS

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### 1. Introduction

The theoretical studies have established that the resistance at oscillations of a rigid body it is not proportional to the body speed in the fluid, as it is considered in the classical methods and in the literature for a convenient calculation.

The resistance force to the ship hull oscillations is depending by the square of the speed, being given by the relation:

$$F_R = 0.5 \cdot \rho_F \cdot A_C \cdot C_D \cdot V_F \cdot |V_F|$$

where:

- $\rho_F$  – fluid density;
- $A_C$  – the area of the projection of the body moving in the fluid on a plane perpendicular to the direction of movement of the fluid;
- $C_D$  – drag coefficient which depends on the shape of the body moving in the fluid;
- $V_F$  – the relative velocity between fluid and body.

In addition to these considerations, the distribution of the wave pressures along the ship length continuously changes, as a function which can be considered to vary cosine-like, by following the wave profile or linear or be constantly, on short time intervals. At the same time, the ship's oscillations also contribute to the change of pressures distributions along the ship length so that the ship's dynamics on the waves is actually

complex and non-linear, but it can be mathematically modeled on short time intervals.

Although the system of the differential equations of second order is nonlinear, it can be assumed that, on small time intervals, the additional masses, the damping and the Smith effect are linear, so that this system is also linear at such intervals, and its stabilized solution varies according to a cosine rule as those of the external loads.

The non-linear system solving is performed by the successive approximation method, in a version adapted to the shape of this system.

The successive approximation method has the disadvantage that in the resonance zones it becomes divergent, and in this case the  $\beta$ -Newmark time integration method is used.

According to this method, the oscillation period is divided in  $N_T$  sufficiently short intervals of  $\Delta t$  duration, delimited by the times, on which the acceleration is considered to be constant and equal to the average of the values from the ends of these intervals.

Based on the numerical method presentation, a code, named RLD-V2, has been developed. The code allows determining of the ship's oscillation parameters as well as the additional sectional efforts in its hull induced by the head wave, considering quadratic damping.

The verification of the calculation method and the developed code was performed by comparing the results of the calculations with the measurements made on the test model in Mejiro test basin, presented in the paper [3].

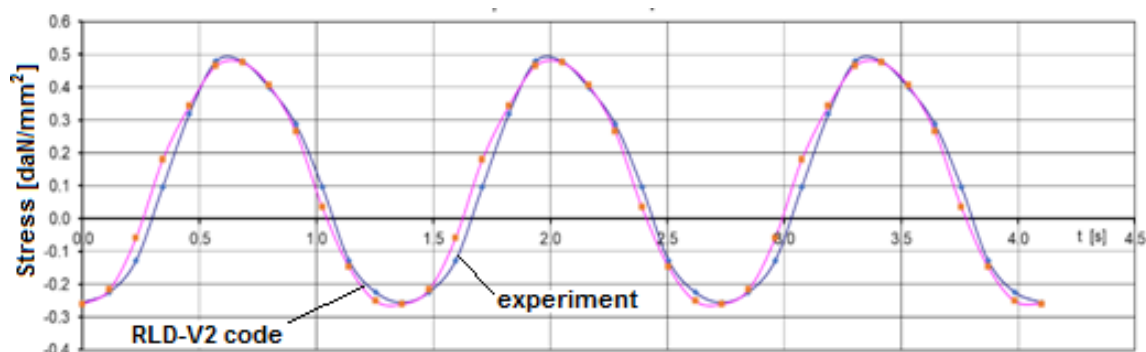


Fig. 1. The time variation of the stress in deck, for the model's speed of 1.39 m/s

The verification of the method and the code was performed by direct analysis of the over time variation of the deck stress at the middle of the model.

In figure 1, the time variation of the stress in deck, for the model's speed of 1.39 m/s is illustrated.

## 2. Conclusions

According to the calculus, as well as measurements from the Mejiro model, it is found that the deviations between the results obtained with the three programs are small, with the indication that the results of the RLD-V2 program are closer to the measurements,

The calculations indicate that up to  $F_n = 0.20$ , it is sufficient to verify the longitudinal strength of the ship to the action of the waves, only by quasi-static laying, which is simpler to apply.

## Acknowledgements

The paper has been supported by Doctoral School of Mechanical and Industrial Engineering from University Dunarea de Jos of Galati, Romania.

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## SHIPS'S HULL STRUCTURAL SAFETY ESTIMATION

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### 1. Introduction

Determination of the floating parameters for the equilibrium position of the ship in still-water and for the quasi-static layout on the wave for different cases of operational loading conditions, as well as the sectional efforts and the elastic line of its hull are the most important analyses aimed to assure the structural safety of the ship hull.

In the literature these analysis is performed principally according to the classical method from papers [1] and [2]. In this paper the author developed and completed in a specific manner, this method to the level of detail required for the programming of the calculation. The theoretical studies have established that the resistance at oscillations of a rigid body it is not proportional to the body speed in the fluid, as it is considered in the classical methods and in the literature for a convenient calculation.

By isolating of an elementary beam "i" from the ship's hull (figure 1) and bringing together the equations of equilibrium of the loads and of the sectional efforts, in the conventional signs of ship system, as well as the relationships between the deformations and the efforts for this beam, is obtained the matrix equation which establishes the relations between the efforts and the deformations at the ends of that elementary beam. Finally, by applying matrix transfer method, the relationship between the aft and forward efforts and deformations is obtained:

$$\bar{v}_n = [D_{n-1}] \cdot \bar{v}_1 + \bar{P}_{n-1} \quad (1)$$

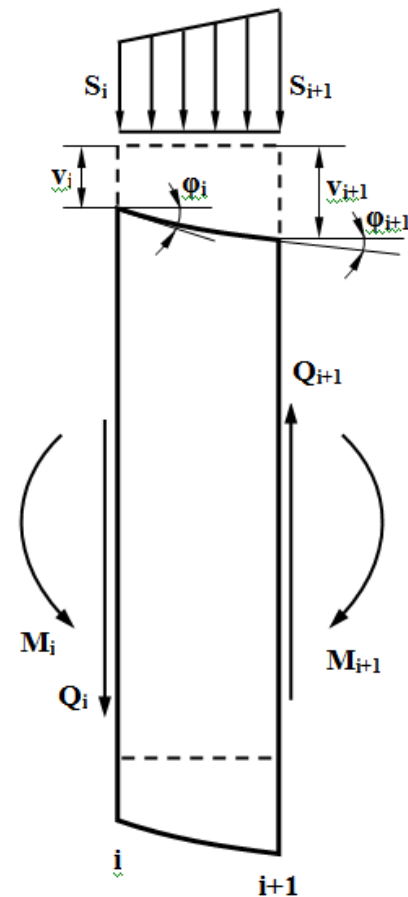
The ends of the ship being free, the sectional efforts are null at the "1" and "n" couples:

$$Q_1 = 0; M_1 = 0; Q_n = 0; M_n = 0. \quad (2)$$

There are 4 deformations and rotations of the ship's ends based on which the elastic line of the hull

can be determined and for their calculation, there are only the last two equations of the equation (1), so that in the first stage, it is considered that the ship's deflections at the ends are zero:

$$v_1 = 0; v_n = 0. \quad (3)$$



**Fig. 1.** Loads, efforts and deformation in an elementary ship hull beam

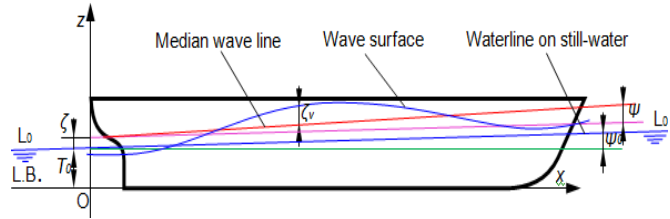
From the  $v_n = 0$  condition, the aft end rotation,  $\phi_1$  is determined and then, by applying the relations (1), the sectional efforts and deformations of the hull along the ship length are determined.

The method is taking into account the ship deformation, so that with the new values of the draughts of the deformed ship, the bending

moments and the shear forces are determined, and then the new elastic line of the ship hull. This numerical procedure is sequential repeated until, in two consecutive iterations, the maximum bending moments differ by less than 0.1%.

Based on the presented calculation method, the RLS-V1 code soft has been developed, whose code being written in the Visual-FORTRAN language that can be run on 32 or 64-bit computers running Windows XP operating system or a later version.

The verification of the code was carried out with a parallelepiped barge, with a uniform distribution mass along its length of 100t/m.



**Fig. 2.** Dynamic layout of the ship on wave

Comparative calculations to verify the RLS-V1 soft for cosine wave are given in Table 1, showing that the differences between manually determined values and program results are below 1%.

**Table 1.** Comparative calculations to verify the RLS-V1 program for layout on cosine wave

| Equilibrium parameters, additional efforts induced by cosine waves, deflection | Analytically determined values |               | Numerically determined values |               |
|--------------------------------------------------------------------------------|--------------------------------|---------------|-------------------------------|---------------|
|                                                                                | On the crest                   | On the hollow | On the crest                  | On the hollow |
| $T_o$ [m]                                                                      | 4.878                          | 4.878         | 4.878                         | 4.878         |
| $\psi$ [rad]                                                                   | 0.000                          | 0.000         | 0.000                         | 0.000         |
| $\theta$ [rad]                                                                 | 0                              | 0             | 0                             | 0             |
| $Q_{wv}(+)$ [kN]                                                               | 12681                          | 12681         | 12675                         | 12675         |
| $Q_{wv}(-)$ [kN]                                                               | -12681                         | -12681        | -12675                        | -12675        |
| $M_{wv}$ [kNm]                                                                 | 403858                         | -403858       | 403449                        | -403448       |
| Deflection [mm]                                                                | 86.00                          | -86.00        | 85.15                         | -85.15        |

## 2. Conclusions

The RLS-V1 code soft for calculating the general section efforts and the elastic line of the ship's hull, at static layout on still-water and at quasi-static layout on the cosine or trochoidal head wave represent a useful tool for design and research activities to improve the construction safety of the ships. The code and method introduce several innovative ideas as follows:

- determination of the elastic line of the hull and its influence on reduction of the bending sectional efforts by applying the transfer matrix method and by using finite macro-elements;
- determining the reduction of bending sectional efforts due to the influence of the elastic line of the hull on the draughts along the ship length;
- determining of the sectional efforts in emergency situations.

## Acknowledgements

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# FINITE ELEMENT STUDY ON THE STRESS STATE ACROSS THE THICKNESS OF A CURVED LAMINATE

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## 1. Introduction

As widely known, an accurate analysis approach of the stress state across the thickness of a composite laminate requires the calculation of the interlaminar through-the-thickness stresses as well as the shear stress components at the layers' interfaces. Moreover, these so-called out-of-plane stresses are the key parameters when defining the delamination strength of the curved laminates, particularly when their radius of curvature and the thickness of the composite are of the same size order. Within the structural design process, when the finite element analysis is carried out to study the stress state of laminate composites, even the three-dimensional models built by layered solid elements allow the calculation of the interlaminar stress components, quite often their practical use is far from a reasonable approach due to the long computational time involved as well as the necessity to perform a careful and well-controlled discretization to avoid the dependencies of the results. In fact, these are the main reasons why the finite element models used for structural stress analysis of lightweight composite structures are generally based on layered shell elements in different depth of discretization. On the other hand, the limitations provided through the theoretical definition of layered shell elements make them impracticable for the study of delamination failure caused by the interlaminar stresses that occur across the thicknesses of the curved laminates.

Several attempts exist in the literature to determine the closed-form analytical solutions for the stress state components developed across the layered curved regions of laminate composite beams. However, their practical use as given in the original analytical form is quite difficult mainly due to various geometries and different loading application points. Lekhnitskii [1], developed the first rigorous solution for analyzing unidirectional laminates under bending by using the theory of elasticity applied to homogeneous orthotropic

materials. Ko and Jackson [2], extended the exact solution to handle different layups under the combined loading of bending, shear and axial force. Closed-form 2-D solutions for the out-of-plane stresses in curved laminated orthotropic beams subjected to shear loads, moments, and temperature changes were developed by Peck [3].

Based on the finite element analysis results, the aim of this paper is to evaluate the stress state originated across the thickness of the curved laminate region of a composite angle under the combined action of axial force, shear force and bending moment, as shown in Fig. 1.

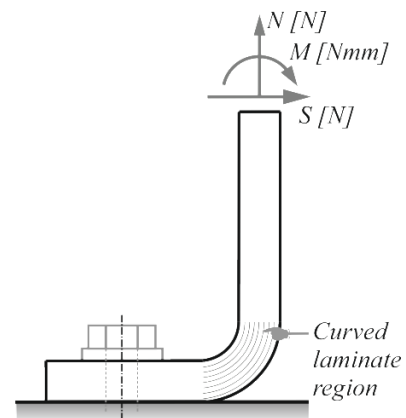


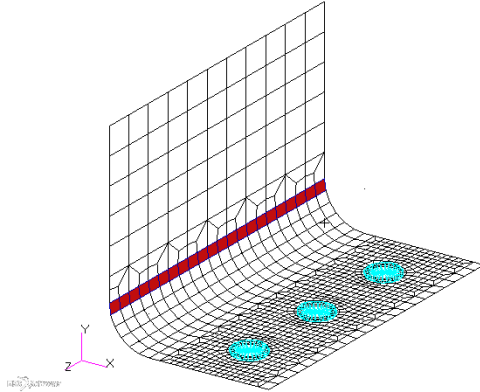
Fig. 1. Typical loading of a laminate curved angle

A simplified calculation approach based on the force and moment fluxes obtained from the results of shell-based finite element models used together with the analytical elasticity solutions provided by the references [2] and [3], is proposed.

## 2. Materials and methodology

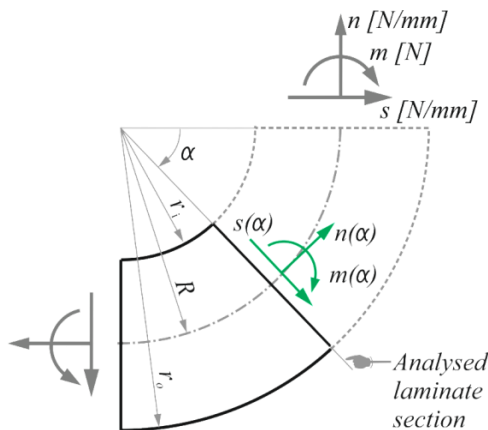
The method is applicable to both symmetrical and nonsymmetrical layups and provides a comprehensive approach by using the 2D finite element analyses of curved laminates assuming that their properties and boundary conditions are constant along the width. It requires the input of shell forces fluxes as well as the moment fluxes extracted at the base line of layered shell elements defining the curved laminate region, as highlighted

in Fig. 2. The analyses were carried out on a finite element model of a laminate curved angle made of PPS-C which follows the stack sequence of  $[(0/90), (+45/-45)_2, (0/90)]_4$  resulting in a thickness of 5 mm and a curvature radius of 10 mm.



**Fig. 2.** The base line of finite elements defining the curved laminate region

The complete 2D finite element analysis was developed in MSC Patran/Nastran [4]. An external loading given by a unit tensile force, a unit shear force and a unit bending moment applied through the upper edge nodes is considered. The boundary conditions were applied at the lower bolted joint by means of rigid body elements as shown in Fig. 2.



**Fig. 3.** The sectional force and moment fluxes definition at an arbitrary section on the curved region

In order to apply the analytical elasticity solutions on different cross sections of the curved laminate region, the shell forces and moments at an arbitrary section defined by the angle  $\alpha$  are derived through the translation of the loads extracted at the base line of shell elements at the beginning of the curved region (see also Fig. 3):

$$n(\alpha) = n \cdot \cos(\alpha) + s \cdot \sin(\alpha) \quad (1)$$

$$s(\alpha) = s \cdot \cos(\alpha) - n \cdot \sin(\alpha) \quad (2)$$

$$m(\alpha) = m + R\{s \cdot \sin(\alpha) - n \cdot [1 - \cos(\alpha)]\} \quad (3)$$

A benchmarking analysis to provide a critical comparison between the results obtained based on a complete three dimensional model and the simplified model build-up by means of layered shell elements is considered too.

### 3. Results

The analysis of results show that the extracted fluxes are locally concentrated due to presence of free edges and bolted joints modelled by rigid elements. However, as the stress state is considered two-dimensional, only normal  $\sigma_r$  and shear stresses  $\tau_{r\theta}$  are used for the interlaminar failure analysis. To obtain reliable results, the proposed calculation method was applied by considering an increment angle  $\alpha$ , equal to  $10^\circ$ . For the prediction of delamination onset, the Kim and Soni [5] failure criterion was applied at each increment.

### 4. Conclusions

The method has the advantage that it allows the use of shell-based finite element results in combination with the analytical closed-form solutions to evaluate the interlaminar stresses generated across the thickness of curved laminates under complex loading conditions. It also shows a good agreement with the solution of three dimensional finite element models and overcomes the problems related to their numerical expensive computations.

### Acknowledgements

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## AUTHOR INDEX

### A

ALEXOPOULOS, Nikolaos..... 19  
ANDRUSZKO, Jakub..... 35, 169  
AVRIGEAN, Eugen ..... 95, 173

### B

BABA, Marius Nicolae..... 207  
BACHRATHY, Daniel ..... 29, 161, 165, 167  
BAGAVAC, Petra ..... 59, 63  
BAGHERIFARD, Sara ..... 87  
BĂIAȘU, Mihai-Liviu ..... 151  
BARCAN (SAVIN), Mihaela..... 203, 205  
BARTHOLMAI, Matthias ..... 139, 141  
BEREZVAI, Szabolcs..... 29  
BERTOCCHI, Enrico ..... 33  
BEZNEA, Elena-Felicia..... 89, 203  
BINDER, Eva..... 153  
BLEICHER, Friedrich ..... 103, 155  
BOGNÁR, Levente László..... 133  
BOLCU, Alexandru..... 91, 93  
BOLCU, Dumitru ..... 91, 93  
BONDREA, Ioan..... 183, 185  
BORBÁS, Lajos ..... 49, 133  
BOUTERF, Amine ..... 53  
BRATU, Polidor..... 187, 189  
BRIA, Vasile ..... 105  
BRODECKI, Adam ..... 145  
BURADA, Cristian Oliviu ..... 91, 93  
BUT, Adrian..... 143

### C

CANARACHE, Radu ..... 143  
CARAMATESCU, Adrian..... 171  
CERNESCU, Anghel..... 25  
CHAHARDEHINEJAD, Omid ..... 141  
CHIRICĂ, Ionel ..... 89, 203, 205  
CHRYSANTHOPOULOU, Eleftheria..... 111  
CIOBÎRCĂ, Cătălin..... 129  
CÎRCIUMARU, Adrian ..... 105  
CONSTANTINESCU, Cătălin..... 129  
CONSTANTINESCU, Dan Mihai..... 9, 21, 39  
COSMI, Francesca..... 113, 127  
COSOIU, Costin Ioan ..... 181  
COSTESCU, Dan Mihail ..... 147, 175  
CRNOJEVIĆ, Stevan..... 41  
CRUCIAT, Radu Iuliu ..... 135, 181  
CSISZÉR, Tamás..... 133  
CZMOCHOWSKI, Jerzy ..... 199

### D

DAL MASO, Alberto ..... 127  
DAMLJANOVIĆ, Dijana ..... 137  
DEGERATU, Mircea..... 181  
DERLUKIEWICZ, Damian ..... 35, 169  
DIŠIĆ, Aleksandar..... 101  
DOBRESCU, Cornelia-Florentina..... 191  
DOMAZET, Željko ..... 59, 63  
DOMBOVARI, Zoltan..... 31  
DOUMOUCHTSIS, Stergios ..... 111  
DRĂGHICI, Marian Sebastian..... 105  
DREXEL, Martin ..... 51  
DZIAŁAK, Paulina..... 37, 149

### E

EBERHARDSTEINER, Josef ..... 61  
ERDMANN, Jessica ..... 159

### F

FAMAHEY, Nele..... 115  
FEUCHTER, Michael..... 39  
FICZERE, Peter..... 49  
FISCHER, Michael..... 23  
FREDDI, Alessandro ..... 5  
FURUTA, Tadahiko ..... 57

### G

GAJDOŠ, Ľubomír ..... 15  
GEONEA, Ionuț Daniel ..... 45, 109  
GHINDEA, Cristian Lucian ..... 135, 181  
GÖBEL, Luise..... 75  
GOLASIŃSKI, Karol ..... 57  
GRAUR, Iulia..... 105  
GRIGORE, Andrei-Mădălin..... 151  
GRUIONU, Gabriel..... 129  
GRUIONU, Lucian Gheorghe..... 129  
GRÜNDER, Klaus-Peter..... 11  
GSELL, Willy ..... 115  
GUAGLIANO, Mario ..... 87  
GULYÁS, Anna ..... 123

### H

HACKSTEINER, Matthias ..... 155  
HADĂR, Anton..... 177, 179  
HAJDU, David ..... 161

|                            |          |
|----------------------------|----------|
| HALÁSZ, Marianna .....     | 123      |
| HASSLINGER, Patricia ..... | 27       |
| HAUSER, Stephan .....      | 11       |
| HECZKO, Jan .....          | 79       |
| HELLMICH, Christian .....  | 27, 71   |
| HELMERICH, Rosemarie ..... | 11       |
| HILD, François .....       | 53       |
| HILLE, Falk .....          | 11       |
| HIMMELREICH, Uwe .....     | 115      |
| HOFKO, Bernhard .....      | 69       |
| HOFSTETTER, Günter .....   | 51       |
| HOLZNER, Armin .....       | 71       |
| HÜLLMANN, Dino .....       | 139, 159 |

## I

|                        |                 |
|------------------------|-----------------|
| IACOB, Ionel .....     | 89              |
| IHLEMANN, Jörn .....   | 13, 81, 83, 157 |
| INSPERGER, Tamás ..... | 125, 161        |
| IONICĂ, Valeriu .....  | 93              |
| ISAKOVIĆ, Jovan .....  | 137             |

## J

|                        |     |
|------------------------|-----|
| JAGSCH, Valentin ..... | 27  |
| JOHANN, Sergej .....   | 141 |
| JONKERS, Ilse .....    | 115 |
| JOVIČIĆ, Gordana ..... | 99  |

## K

|                               |               |
|-------------------------------|---------------|
| KADOKE, Daniel .....          | 11, 23        |
| KANZENBACH, Lars .....        | 83            |
| KARLIŃSKI, Jacek .....        | 149           |
| KEMÉNY, Dávid Miklós .....    | 163           |
| KISS, Adam .....              | 165           |
| KISS, Bernadett .....         | 119, 121      |
| KISS, Rita .....              | 119, 121, 123 |
| KLINGENBRUNNER, Martin .....  | 103           |
| KOCÁB, Jiří .....             | 55            |
| KOHLHOFF, Harald .....        | 23, 159       |
| KOLAREVIĆ, Nenad .....        | 41            |
| KÖNIGSBERGER, Markus .....    | 75            |
| KOSSA, Attila .....           | 29, 55, 85    |
| KOTTNER, Radek .....          | 55, 79        |
| KOURKOULIS, Stavros .....     | 19, 111       |
| KOVÁCS, Bálint Ádám .....     | 49            |
| KOVÁCS, Dorina .....          | 163           |
| KOWALEWSKI, Zbigniew .....    | 57, 145       |
| KOWITZ, Astrid .....          | 11            |
| KROUPA, Tomáš .....           | 77, 201       |
| KRSTULOVIC-OPARA, Lovre ..... | 63            |
| KRYSTEK, Jan .....            | 79            |
| KULÍŠEK, Viktor .....         | 65            |
| KURAMOTO, Shigeru .....       | 57            |
| KUZMANOVIĆ, Vladimir .....    | 195           |

## L

|                            |                 |
|----------------------------|-----------------|
| LAHAYNE, Olaf .....        | 27, 61          |
| LAZARIDOU, Ilona .....     | 19              |
| LEHMANN, Thomas .....      | 13, 81, 83, 157 |
| LEITNER, Martin .....      | 73              |
| LI, Chong .....            | 153             |
| LOVASZ, Zoltan .....       | 31              |
| LUCZYNSKI, Krzysztof ..... | 71              |
| LUKŠA, Francisko .....     | 59              |
| LUPAȘCU, Dumitru .....     | 203, 205        |

## M

|                              |         |
|------------------------------|---------|
| MAJ, Michał .....            | 57      |
| MAKOWSKA, Katarzyna .....    | 145     |
| MALCIU, Raluca .....         | 109     |
| MANEA, Marilena .....        | 151     |
| MANG, Herbert .....          | 153     |
| MANTOVANI, Sara .....        | 33      |
| MARGHITU, Dan .....          | 117     |
| MAŚLAK, Paweł .....          | 199     |
| MATYJEWSKI, Marek .....      | 17      |
| MAXIMOVA, Natalia .....      | 113     |
| MICKOVIĆ, Dušan .....        | 41      |
| MILOŠ, Marko .....           | 41, 137 |
| MILOVANOVIĆ, Vladimir .....  | 99, 101 |
| MIRIȚOIU, Cosmin Mihai ..... | 91, 93  |
| MITROVIĆ, Časlav .....       | 43      |
| MOCANU, Costel Iulian .....  | 171     |
| MOCIAN, Oana Alexandra ..... | 39      |
| MOLLA, Mauro Filippo .....   | 87      |
| MOLNÁR, Csenge .....         | 125     |
| MOLNÁR, László .....         | 133     |
| MOLNAR, Tamas .....          | 29      |
| MUNTENIȚĂ, Cristian .....    | 105     |

## N

|                        |          |
|------------------------|----------|
| NEITZEL, Frank .....   | 23       |
| NEU, Corey .....       | 115      |
| NEUMANN, Patrick ..... | 139, 159 |
| NUȚU, Emil .....       | 9        |
| NYIRŐ, József .....    | 163      |

## O

|                         |         |
|-------------------------|---------|
| OBSZAŃSKI, Marcin ..... | 17      |
| OELSCH, Erik .....      | 83      |
| OLESIK, Mihaela .....   | 95, 173 |
| OLESIK, Valentin .....  | 95, 173 |
| OPACIĆ, Mirjana .....   | 43      |
| OSBURG, Andrea .....    | 75      |

## P

|                        |     |
|------------------------|-----|
| PĂDEANU, Adriana ..... | 91  |
| PÁLYA, Zsófia .....    | 123 |

|                                 |                   |
|---------------------------------|-------------------|
| PASCU, Adrian .....             | 95, 173, 183, 185 |
| PASTRAMĂ, Maria-Ioana .....     | 115               |
| PASTRAMĂ, Ștefan Dan .....      | 129, 147          |
| PĂTRAȘCU, Alexandru Ionuț ..... | 175, 177, 179     |
| PERETZKI, Edgar .....           | 81                |
| PERGIALIOTIS, Vasileios .....   | 111               |
| PERREA, Despina .....           | 111               |
| PETCU, Alin .....               | 109, 117          |
| PETRO, Balint .....             | 121               |
| PETROVIĆ, Nebojša .....         | 43                |
| PETRUSE, Radu Emanuil .....     | 183, 185          |
| PICHLER, Bernhard .....         | 61, 75, 153       |
| PIECZYSKA, Elżbieta .....       | 57                |
| PÎRVULESCU, Liviu Daniel .....  | 25                |
| PLACHY, Robert .....            | 71                |
| PLATZER, Julia .....            | 103               |
| POMBERGER, Sebastian .....      | 73                |
| POPA, Sorin .....               | 189, 193          |
| POŠVÁŘ, Zdeněk .....            | 65                |
| PUȘCAȘU, Samuel .....           | 183, 185          |
| PYRZANOWSKI, Paweł .....        | 67                |

## R

|                            |          |
|----------------------------|----------|
| RĂCĂNEL, Ionuț Radu .....  | 135, 181 |
| REIHSNER, Roland .....     | 61       |
| RENKÓ, József Bálint ..... | 163      |
| ROȘCA, Nicolae .....       | 95, 173  |
| ROȘU, Dorin .....          | 39       |
| RŮŽICKA, Milan .....       | 65       |

## S

|                                |           |
|--------------------------------|-----------|
| SANDU, Adriana .....           | 9, 21     |
| SANDU, Diana-Ionela .....      | 151       |
| SANDU, Marin .....             | 9, 21, 39 |
| SAVA, Violeta .....            | 105       |
| SCHEINER, Stefan .....         | 27, 71    |
| SCHMID, Sophie .....           | 61        |
| SCHWAIGER, Clemens .....       | 155       |
| SCHWARZINGER, Henrik .....     | 11        |
| SEBE, Cristinel .....          | 193       |
| SMANIOTTO, Stefan .....        | 51        |
| SMOLÍK, Jan .....              | 65        |
| SMOLNICKI, Tadeusz .....       | 199       |
| ȘOIMU-IACOB, Andreea .....     | 129       |
| SOROHAN, Ștefan .....          | 9, 21, 39 |
| ȘPERL, Martin .....            | 15        |
| SPILIOPOULOU, Chara .....      | 111       |
| STAŢCO, Mariusz .....          | 35, 37    |
| STĂNESCU, Marius Marinel ..... | 91, 93    |
| STANKOVIĆ, Miloš .....         | 41        |

|                             |                  |
|-----------------------------|------------------|
| STASZCZAK, Maria .....      | 57               |
| STEINER, Daniel .....       | 69               |
| STEPAN, Gabor .....         | 29, 85, 161, 165 |
| STEPNOWSKI, Marek .....     | 67               |
| STOCKMANN, Martin .....     | 13, 81, 157      |
| STOICESCU, Liviu-Dan .....  | 205              |
| STOSCHKA, Michael .....     | 73               |
| STRANGFELD, Christoph ..... | 141              |
| STROZZI, Antonio .....      | 33               |
| SURMA, Robert .....         | 53               |
| SZYMCZAK, Tadeusz .....     | 145              |

## T

|                           |          |
|---------------------------|----------|
| TARNIȚĂ, Daniela .....    | 109, 117 |
| TODIĆ, Ivana .....        | 195      |
| TOMIČEVIĆ, Zvonimir ..... | 53       |
| TONCIU, Oana .....        | 97, 191  |

## U

|                              |     |
|------------------------------|-----|
| UDRIȘTOIU, Anca .....        | 129 |
| URDAREANU, Vlad Daniel ..... | 181 |

## V

|                        |              |
|------------------------|--------------|
| VASILE, Ovidiu .....   | 97, 187, 189 |
| VOROTOVIĆ, Goran ..... | 43           |
| VRBA, Pavel .....      | 65           |
| VUKOVIĆ, Đorđe .....   | 137          |

## W

|                       |        |
|-----------------------|--------|
| WEISBRICH, Sven ..... | 23     |
| WU, Cheng-Chieh ..... | 11, 23 |

## Y

|                  |     |
|------------------|-----|
| YUAN, Yong ..... | 153 |
|------------------|-----|

## Z

|                                  |                    |
|----------------------------------|--------------------|
| ZAHARIA, Cătălin .....           | 187, 193           |
| ZAHARIA, Nicușor Laurențiu ..... | 147, 175, 177, 179 |
| ZELEI, Ambrus .....              | 125                |
| ZEMČÍK, Hana .....               | 77, 201            |
| ZEMČÍK, Robert .....             | 77, 201            |
| ZEVENBERGEN, Lianne .....        | 115                |
| ZIDARU, Elvis-Georgian .....     | 147, 177, 179      |
| ŽIVKOVIĆ, Miroslav .....         | 99, 101            |