ASMO-UK

Engineering Design Optimization

Product and Process improvement

Extended Abstracts of the 10th ASMO UK / ISSMO conference

Delft University of Technology
Delft, The Netherlands
June 30th – July 1st, 2014

TU Delft Delft University of Technology

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Association for Structural and Multidisciplinary Optimization in the UK

UK Branch of ISSMO - International Society of Structural and

10th ASMO UK / ISSMO conference on Engineering Design Optimization
Delft University of Technology, Delft, The Netherlands,
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Monday 30th June 2014	
9:10 - 9:20	**Welcome address by Prof.dr. Theun Baller, Dean of the Faculty Mechanical, Maritime and Materials Engineering**
9:20 - 9:30	**ASMO-UK 10 - Opening & Welcome Address: by Prof. Vassili V. Toropov (ASMO-UK) and Prof.dr.ir. Fred van Keulen (TUD Host)**
Session 1	**Session 1:** ***Advanced methods and techniques (sensitivities, meta models, noise, GPU, isogeometric)*** ***Chaired by: Dr.ir. Matthijs Langelaar***
9:30 - 10:00	Franz-Joseph Barthold, **Nikolai Gerzen**, *Design Space Exploration Based on the Analysis of the Inner Structure of Sensitivities*
10:00 - 10:30	**Greg N. de Boer**, R. W. Hewson, H. M. Thompson, L. Gao, V. V. Toropov, *Metamodelling and Optimisation in Two-Scale Elastohydrodynamic Lubrication*
10:30 - 11:00	**M. H. Aissa** and Dr. T. Verstraete, *Use of modern modern GP Us on Design Optimization*
11:00 - 11:30	**Coffee**
Session 2	**Session 2:** ***Advanced methods and techniques (sensitivities, meta models, noise, GPU, isogeometric)*** ***Chaired by: Dr. Osvaldo M. Querin***
11:30 - 12:00	**C. A. Gilkeson**, V. V. Toropov, H. M. Thompson, M. C. T. Wilson, N. A. Foxley and P. H. Gaskell, *The Curse of Numerical Noise and Implications for CFD-Based Design Optimization*
12:00 - 12:30	Etienne Lemaire, Samih Zein, Michaël Bruyneel and **Pierre Duysinx,** *Optimal design of composite structures with curved fiber trajectories*
12:30 - 13:00	**Zhen-Pei Wang** and Sergio R. Turteltaub, *Isogeometric Shape Optimization for Quasi-Static Mechanical Problems*
13:00 - 14:00	**Lunch**
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14:00 - 14:30	**Andrew Shires**, *Optimisation of a MW scale offshore vertical axis wind turbine*
14:30 - 15:00	**Igor N. Egorov**, Gennany V. Kretinin, Koystantin S. Fedechkin, *Multi-Objective Optimization of Real-Life Complex Objects Using IOSO Technology*
15:00 - 15:30	**Silvia Poles**, *Multiobjective and Robust Design Optimization of an Active Solar Energy System*
15:30 - 16:00	**Coffee & Conference Photograph**
Session 4	**Session 4:** ***Multiobjective and/or Multi-Disciplinary Optimization*** ***Chaired by: Prof. Pierre Duysinx***
16:00 - 16:30	T. Wilschut, A.T. Hofkamp, **L.F.P. Etman**, J.E. Rooda, and A.W.J. Gielen, *Specifying MDO coupling structure using the Psi language: a LED lighting design case*
16:30 - 17:00	**Jonathan Ollar**, Vassili Toropov, and Royston Jones, *Mid-Range Approximations in Sub-spaces for MDO problems including disparate attribute models*
17:00 - 17:30	**C. Chahine** and T. Verstraete, *Multidisciplinary Optimization of a Transonic Fan Blade for High Bypass Ratio Turbofan Engines*
Session 5	**Session 5:** ***Topology Optimization*** ***Chaired by: Prof. George I.N. Rozvany***
17:30 - 18:00	Stephan Hunkeler and **Fabian Duddeck**, *Optimal Topologies of Extrusion Cross-sections for Crashworthiness*
18:00 - 18:30	**Osvaldo M. Querin**, Mariano Victoria, Concepción Díaz and Pascual Martí, *Study of stiffener design for the Pantheon's dome using the Isolines Topology Design method*
18:30 - 19:00	**Evert Hooijkamp** and Fred van Keulen, *Reduced Modal-Basis Techniques in Transient Thermo-Mechanical Topology Optimization.*
19:00 - 19:30	**Walk to the Restaurant-Café In de Waag**
19:30 - 20:00	**Pre-drinks reception**
20:00	**Conference Dinner Starts**

Tuesday 1st July 2014	
Session 6	**Session 6:** ***Topology Optimization*** ***Chaired by: Vassili V. Toropov***
9:00 - 9:45	**George Rozvany** · Vanda Pomezanski · Tomasz Sokół, *Sense and nonsense in structural topology optimization*
9:45 - 10:15	C. Alonso, R. Ansola and E. Veguería, **O.M. Querin**, *A MATLAB code for structural and compliant topology optimization with the Sequential Element Rejection and Admission method*
10:15 - 10:45	**Fred van Keulen,** Gijs van der Veen, Evert Hooijkamp, Matthijs Langelaar, *Additive Manufacturing and Topology Optimization -- Symbiosis and Challenges*
10:45 - 11:00	**Coffee**
Session 7	**Session 7:** ***Composites*** ***Chaired by: Fred van Keulen***
11:00 - 11:30	**Anna Arsenyeva** and Fabian Duddeck, *Iso-contour method for optimization of steered-fiber composites*
11:30 - 12:00	**J. Chesterfield**, A. T. Rhead, R. Butler, *Optimization of Curved Fibre Composites for Buckling Performance with Manufacturing Constraints*
12:00 - 12:30	**Edgars Labans**, Kaspars Kalnins, Philippe Lefort, *Weight optimisation of the truck cabin part from thermoplastic composites*
12:30 - 12:40	**Conference Close**
12:40 - 14:00	**Lunch**

Use of modern GPUs on Design Optimization

M. H. Aissa* and Dr. T. Verstraete†
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Graphics Processing Units (GPUs) are a promising alternative hardware to Central Processing Units (CPU) for accelerating applications with a high computational power demand. In many fields researchers are taking advantage of the high computational power present on GPU to speed up their applications. These applications span from data mining to machine learning and life sciences. The field of design optimization has been also influenced by this alternative hardware. The automated search on the design space has been delegated to GPUs or to a system of CPUS assisted by GPUs. This paper is among the firsts to analyze the use of GPUs especially on design optimization. The focus was on topology optimization, shape optimization and multidisciplinary design optimization (MDO). The target is to highlights not only the progress done on running optimization methods on GPU but also the limitations that researchers have to cope with and the areas that require more research.

Nomenclature

CPU	=	Central Processing Unit
HPC	=	High Performance Computing
GPU	=	Graphics Processing Unit
MDO	=	Multidisciplinary Design Optimization
MPI	=	Message Passing Interface

I. Introduction

Design engineers are interested on having a design with an optimal performance under specified constraints. The problem is then formulated using an objective function or multiple ones. The objective function has to be minimized under a set of constraints using a design range for the design vector. This approach leads to a computation intensive problem, since the design space is large for relevant problems and the objective function is not always trivial. The optimization algorithm itself is most of the time an iterative process requiring in every iteration the evaluation of a high number of designs.

To tackle this time performance issue, engineers started parallelizing their applications to run them on a high number of processors. This step marks the entrance of design optimization on the field of High Performance Computing (HPC). This is related with an increase of the programming burden, since the designer has to distribute the computational work among the available CPU processors and regulate the communication using MPI.

In addition to HPC solutions, another idea with a practical aspect emerged and is now widely used. The complicated objective function has been replaced by a less complicated model (*metamodel*) that generates the design evaluation on shorter time but with a loss of accuracy. The delicate task is to combine high fidelity (original objective function) and low fidelity (metamodel) evaluations to accelerate the design optimization process and keep the metamodel accurate enough.

The appearance of programmable GPUs enabled an access to a high computational power system different from known CPU-HPC systems. GPUs are indeed a shared-memory system. A large number of cores are sharing the same memory. The programmer is no more responsible for communication and data distribution. These GPU-cores are available in large number and specialized on arithmetic computation unlike the more powerful but general purpose CPU cores. The work of the design engineer is then to successfully divide the global optimization problem on small work packages that can be handled by a GPU core in a massively parallel manner. The problems that are easily divided on small and independent work packages are called *embarrassingly parallel*. If the work packages are not independent and need intercommunication, the problem is called *coarse-grained parallel*. If the communication increases, it is then called *fine-grained parallel*.

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For this work we analyzed the use of GPUs to accelerate optimization methods used in shape optimization, topology optimization and multidisciplinary design optimization. Optimization methods are classified after the order of gradients used: Zero-order methods require only objective function evaluations and no gradients; first-order methods rely on first order gradients and second-order methods make use of the Hessian matrix.
The first section of this paper deals with the domain of application of design optimization covered in this work. The next section emphasizes on the different optimization methods used on the literature and their parallelization potential. The third section discusses the advantages derived from the GPU use and highlights the areas requiring more research toward a better use of GPU high computational power.

II. Conclusion

This paper covered the use of GPU to accelerate design optimization problems focusing on shape optimization, topology optimization and multidisciplinary design optimization. We found, that independently of the domain of application, methods of zero-order such as evolutionary algorithms, ant colony optimization and particle swarm optimization are showing the highest performance gain through GPU, since they exhibit a clearer parallelism and are simpler for rewriting on GPU language. The GPUs are saving in many cases for zero-order methods an important part of computation time. One order of magnitude as overall speedup is a common value.

First- and second-order methods such as steepest descent or conjugate gradients are related to a higher need of memory access leading to a reduced GPU performance gain.

Concerning multidisciplinary design optimization we observed a rarer use of GPUs, which is basically influenced by the higher complexity of MDO problems and the multitude of tools and simulation needed in comparison with standard design optimization.

III. Acknowledgments

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A matlab code for structural and compliant topology optimization with the Sequential Element Rejection and Admission method

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University of Leeds, Leeds, LS2 9JT, United Kingdom

Topology optimization techniques have provided complementary tools for multidisciplinary design. A wide variety of methods have been developed since the first method was presented. Most topology optimization techniques have provided a simplified matlab code for explanatory and educational purposes. The Sequential Element Rejection and Admission (SERA) method was proposed by Rozvany and Querin and has now been extended to structural and compliant mechanisms design. Research has shown that the method is robust and versatile in providing optimized topologies. The purpose of this paper is therefore to provide a simplified 125 lines matlab code to design structures and compliant mechanisms with the SERA method.

Nomenclature

ρ_e	=	density of the e^{th} finite element
ρ_{min}	=	minimum density
N	=	number of finite elements
MPE	=	Mutual Potential Energy
$\boldsymbol{K}$	=	global stiffness matrix
$\boldsymbol{F_1}$	=	nodal force vector containing the input thermal load
$\boldsymbol{F_2}$	=	nodal force vector containing the unit output force
$\boldsymbol{U_1}$	=	displacement field due to load case 1
$\boldsymbol{U_2}$	=	displacement field due to load case 2
k_{out}	=	output stiffness
$\boldsymbol{F_{1e}}$	=	nodal force vector containing the input thermal load of the e^{th} finite element
$\boldsymbol{K_e}$	=	elemental stiffness matrix
$\Delta V_{Remove}(i)$	=	volume to be removed in the i^{th} iteration
$\Delta V_{Add}(i)$	=	volume to be added in the i^{th} iteration
α_e	=	elemental sensitivity number
$\boldsymbol{\alpha}_R$	=	vector of sensitivity numbers related to real material
$\boldsymbol{\alpha}_V$	=	vector of sensitivity numbers related to virtual material
α^{th}_R	=	threshold value of real material
α^{th}_V	=	threshold value of virtual material
ε_i	=	convergence criteria

I. Introduction

Most of the widely used topology optimization methods have presented a simplified Matlab code for researchers and students to try and analyze the different methods. Matlab, as a high-level programming language that allows for the solution of scientific problems with minimum coding effort, is the perfect platform to present new topology optimization methods. In addition, most of the new methods that have presented a

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Matlab code have based their programming on the first method to present such an educational support, the SIMP method[2] and Sigmund's 99 line topology optimization code[1]. Straightforward and accessible syntax allows Matlab to perfectly fulfill the educational purposes of these proposals. Other examples of Matlab codes for methods such as, among others, the Level Set method[7] or the Pareto optimal tracing[6] can be found in the literature.

The Sequential Element Rejection and Admission (SERA) method was proposed in 2002 by Rozvany and Querin[3] and has now been extended to structural and compliant mechanisms design by the authors[4,5]. Research has shown that the method is robust and versatile in providing optimized topologies. The purpose of this paper is therefore to provide a Matlab code to design structures and compliant mechanisms with the SERA method for educational purposes.

For the sake of simplicity as it is a well known code in the research community, the code has been developed with the 99 line code presented by Sigmund[1] as a starting point. Two benchmark examples are used to show the validity of the programming code proposed.

II. The generalized SERA method

The SERA method is bi-directional in nature and considers two separate material models: 1) real material and 2) a virtual material with negligible stiffness[3]. Two separate criteria of rejection and admission of elements allow material to be introduced and removed from the design domain by changing its status from virtual to real and vice versa (Figure 1). The final topology is constructed from all the real material present at the end of the optimization. The use of discrete variables guarantees that the final topology will be free of gray areas as it can be found in methods with continuous variables such as SIMP.

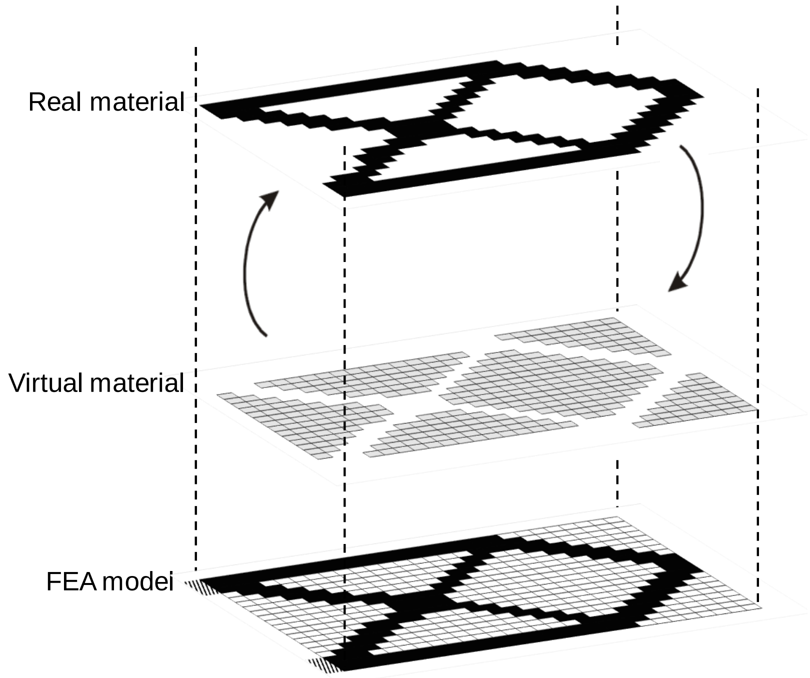

Figure 1: The SERA real and virtual material models in structural design

In this work, the concept of two material models and separate criteria are maintained. The design criterion is generalized to any response parameter that may be considered as the objective function in structural topology optimization.

The twelve steps that drive the SERA method for structural optimization problems are given below, and can be seen in the flow chart of Figure 2.

1) Define the design problem. The maximum design domain must be defined and meshed with finite elements. All boundary constraints, loads and the target volume fraction V^* must also be specified.
2) Assign real and virtual material properties to the initial design domain.
3) Calculate the variation of the volume fraction in the i^{th} iteration which consists of the volume fraction to be added $\Delta V_{Add}(i)$ and removed $\Delta V_{Remove}(i)$.
4) Carry out a Finite Element Analysis (FEA) of the structure.
5) Calculate the sensitivity number in each element α_e.
6) Apply a mesh independent filtering.

7) Separate the values of the driving criterion, in this case the sensitivity number, in each element into real and virtual materials, α_R and α_V.
8) Define the threshold values for real and virtual material, α_R^{th} and α_V^{th}, that will allow the required volume fraction of material to be removed or added.
9) Remove and add elements.
10) Calculate the volume of the real material in the domain.
11) Calculate the convergence criterion $\mathcal{E}_i$.
12) Repeat steps (3) through (11) until the target volume is reached and the optimization converges. The final topology is represented by the real material in the design domain.

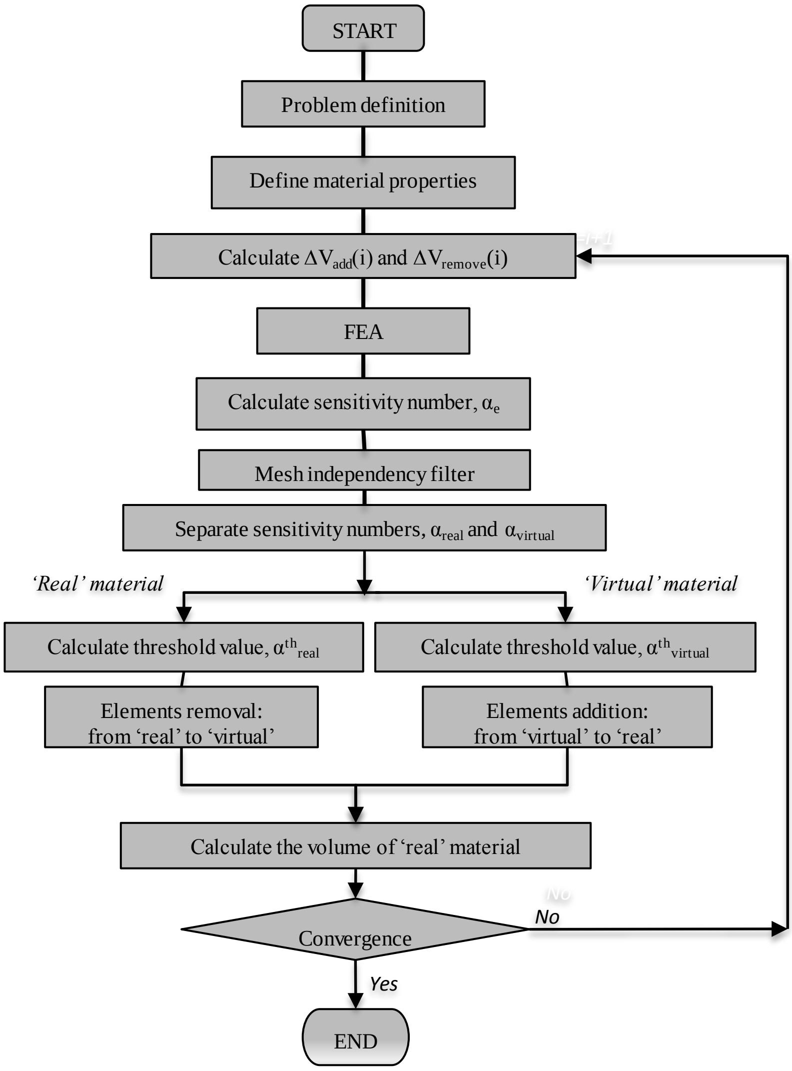

Figure 2: Flow chart of the SERA method

A. Definition of the initial design domain

The SERA method can start from a full design domain (all elements are real material), from a void design domain (all elements are virtual material), with any amount of material present in the domain and with any distribution of the material in the design domain. For any of these cases, the material present in the domain is assigned the real material properties and material not present in the domain is assigned the virtual material properties. The SERA method converges toward the optimum topology regardless of the initial design domain.

B. Calculating the volume to add $\Delta V_{Add}(i)$ and remove $\Delta V_{Remove}(i)$

Material is added and removed from the design domain in a two stage process (See Figure 3 and Figure 4):
1) Different amounts of material are added and removed in each iteration until the target volume fraction V* is reached.
2) Once the target volume fraction is reached, material re-distribution takes place by both adding and removing the same amount of material until the problem converges.

I. Determining the target volume fraction

The target volume fraction of stage 1 depends on the starting design domain and on the iteration number. Two starting domain cases exist: a) when the design starts with a volume fraction higher than the target fraction or a full domain, $V_F(0)$ or b) when the design starts with a volume fraction lower than the target fraction or from a void domain $V_V(0)$. For case (a), the target volume fraction $V_F(i)$ is calculated using (1) and for case (b), the target volume fraction $V_V(i)$ is calculated using (2). The total amount of material to be both added and removed in the i^{th} iteration is the given by (3). This value is then separated into the volume fraction to be added $\Delta V_{Add}(i)$ and the volume fraction to be removed $\Delta V_{Remove}(i)$. For case (a), these terms are given by (4) and (5), Figure 3, and for case (b) they are given by (6) and (7), Figure 4.

$$V_F(i) = \max(V_F(i-1)\cdot(1-PR), V^*) \tag{1}$$
$$V_V(i) = \min(V_V(i-1)+PR, V^*) \tag{2}$$
$$\Delta V(i) = |V_{F\text{ or }V}(i) - V_{F\text{ or }V}(i-1)| \tag{3}$$
$$\Delta V_{Add_F}(i) = \Delta V(i)\cdot(SR-1) \tag{4}$$
$$\Delta V_{Remove_F}(i) = \Delta V(i)\cdot SR \tag{5}$$
$$\Delta V_{Add_V}(i) = \Delta V(i)\cdot SR \tag{6}$$
$$\Delta V_{Remove_V}(i) = \Delta V(i)\cdot(SR-1) \tag{7}$$

where: *PR* is the Progression Rate, with typical values ranging between 0.02-0.06 in structural design; *SRs* is the Smoothing Ratio, with typical values in the range between 1.3 and 1.5 in structural design.

A graphical representation of the removal and addition of elements in each iteration for case (a) is given in Figure 3 and for case (b) in Figure 4. In both cases, each iteration consists of two sub-steps which add and remove material from the design domain. The difference depends on the amount of material to be added or removed so that the volume fraction in that iteration decreases for case (a) (Figure 3) or increases for case (b) (Figure 4).

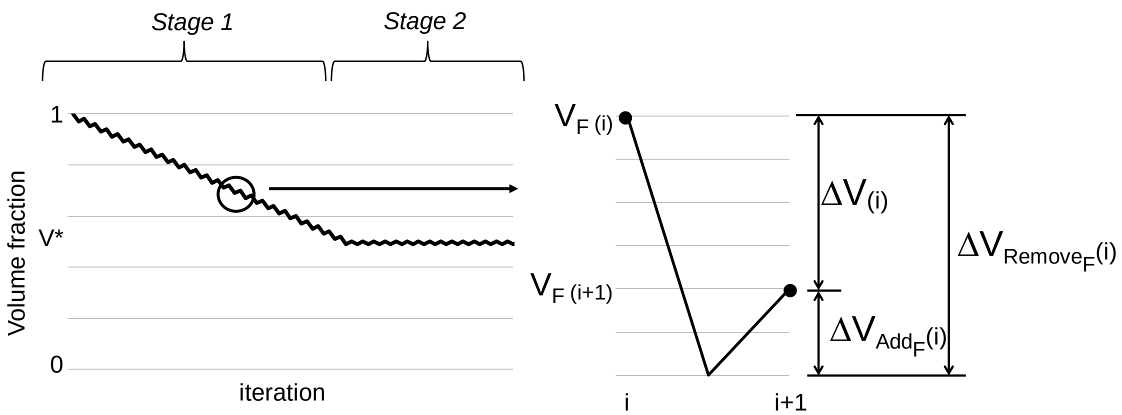

Figure 3: Scheme of the removal and addition of material from a full domain

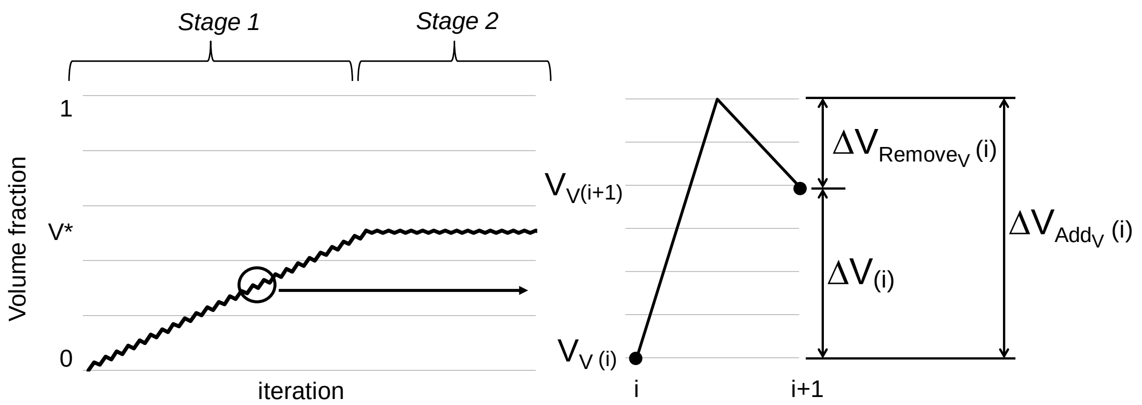

Figure 4: Scheme of the removal and addition of material from a void domain

II. Material re-distribution

The process of material re-distribution consists of both adding and removing the same amount of material from the design domain, as shown in Eq. 8.

$$\Delta V_{Add_{F\ or\ V}}(i) = \Delta V_{Remove_{F\ or\ V}}(i) = \beta \cdot V^* \tag{8}$$

where: β is the material re-distribution fraction, with typical values ranging between 0.001 and 0.005.

C. Removal and addition of elements

The sensitivity number for the e^{th} finite element α_e is a function of the variation between two iterations in the stiffness matrix of that element ΔK_e (9).

$$\Delta \mathbf{K}_e = \mathbf{K}_e(i) - \mathbf{K}_e(i-1) \tag{9}$$

where $K_e(i)$ is the stiffness matrix in the i^{th} iteration for the e^{th} finite element; and $K_e(i\text{-}1)$ is the stiffness matrix in the $(i\text{-}1)^{th}$ iteration for the same finite element.

If an element is added, $\boldsymbol{K}_e(i) = \boldsymbol{K}_e$ and $\boldsymbol{K}_e(i-1) \approx 0$, so the variation of the elemental stiffness matrix is $\Delta\boldsymbol{K}_e = \boldsymbol{K}_e$. But if an element is removed, $\boldsymbol{K}_e(i) \approx 0$ and $\boldsymbol{K}_e(i-1) = \boldsymbol{K}_e$, and $\Delta\boldsymbol{K}_e = -\boldsymbol{K}_e$. The elemental sensitivity number for the real and virtual material is given by (10) and (11), respectively.

$$DC_{eR} = \alpha_{eR} = -\frac{1}{2} \cdot \mathbf{U}_e^T \cdot \mathbf{K_e} \cdot \mathbf{U_e} \tag{10}$$

$$DC_{eV} = \alpha_{eV} = \frac{1}{2} \cdot \mathbf{U}_e^T \cdot \mathbf{K_e} \cdot \mathbf{U_e} \tag{11}$$

As the objective is to minimize the compliance of the design, the elements with the lower values of sensitivity number are the ones to be added and removed.

The threshold values $\alpha_R{}^{th}$ and $\alpha_V{}^{th}$ are the sensitivity values that remove or add the amount of volume $\Delta V_{Remove}(i)$ and $\Delta V_{Add}(i)$ defined for each iteration.

D. Mesh independent filtering

The mesh independent filter proposed in this SERA method is based on the one by Sigmund and Petersson[8] and modifies the elemental Driving Criterion DC_e based on a weighted average of the elemental Driving Criteria (12) in a fixed neighbourhood defined by a minimum radius r_{min} (13).

$$DC_e' = \frac{\sum_{i=1}^{n} \rho_i \cdot \mu_i \cdot DC_i}{\sum_{i=1}^{n} \mu_i} \tag{12}$$

$$\mu_i = r_{min} - dist(e,i), \qquad \{i \in n \,/\, dist(e,i) \le r_{min}\}, \qquad e = 1, \dots, n \tag{13}$$

where: DC'_e is the e^{th} element filtered Driving Criterion; n is the number of elements which are inside of the filter radius; ρ_i is the density of element i; μ_i is the weighting factor for element i; its value decreases linearly the further element i is away from element e and for all elements outside the filter radius its value is equal to zero; DC_i is the i^{th} element Driving Criterion; r_{min} is the filter radius specified by the user; $dist(e,i)$ is the distance between the centres of elements e and i.

E. Convergence criterion

The convergence criterion is defined as the change in the objective function in the last 10 iterations (14), which is considered an adequate number of iterations for the convergence study. It implies that the process will have a minimum of 10 iterations as the convergence criterion is not applied until the iteration number has reached 10.

$$\mathcal{E}_i = \frac{\left|\sum_{i=9}^{i=5} DC_i - \sum_{i=4}^{i} DC_i\right|}{\sum_{i=4}^{i} DC_i} \tag{14}$$

where: $\mathcal{E}_i$ is the convergence criterion, with typical values ranging between 0.001-0.01.

III. Matlab Codes

A. The SERA method for structural topology optimization

The matlab code proposed for the generalized SERA method for structural topology optimization is here presented. The program routine is called with the following command line:

```
SERA_STR(NelX,NelY,VolObj,PR,SR,B,Rmin,VolIni,Xmin,Case)
```

where:

NelX, Number of elements in the X axis (horizontal)
NelY, Number of elements in the Y axis (vertical)
VolObj, Target volume fraction
PR, Progress Ratio
SR, Smoothing Ration
B, Material re-distribution fraction
Rmin, Filtering radius for the filtering technique
VolIni, Initial volume (This code has been simplified to have an initial volume fraction VolIni=1)
Xmin, Minimum density considered, a typical value used is 10^{-4}
Case, Two predefined cases are included: Case='c' for a cantilever; and Case= 'm' for a beam with a force at the centre of the bottom edge

```
%%% SERA method for Structural Top Opt ©CristinaAlonsoGordoa
function SERA_STR(NelX,NelY,VolObj,PR,SR,B,Rmin,VolIni,Xmin,Case)
clc; close all; tic; disp([num2str(toc),'......START.......']);
SenNr=sparse(NelY,NelX); x(1:NelY,1:NelX) = VolIni; i = 1;
   Vol(i)=VolIni; Change = 1.; %inicialization of variables
while Change >  0.001
i = i + 1;
Vol(i)= max(Vol(i-1)*(1-PR),VolObj); %For initial design
domain>VolObj
[U,KE,K]=FE(NelX,NelY,x,Case);
Compl(i)= 0.5*U'*K*U; %Compliance in the ith iteration
for ely = 1:NelY
    for elx = 1:NelX
        n1 = (NelY+1)*(elx-1)+ely;
        n2 = (NelY+1)* elx   +ely;
        Ue = U([2*n1-1;2*n1; 2*n2-1;2*n2; 2*n2+1;2*n2+2;
         2*n1+1;2*n1+2],1);
        SenNr(ely,elx) = 0.5*Ue'*KE*Ue; %Objective: minimize C =max
         (-C)=-(-0.5*U*K*U)
     end
end
[SenNr]=Filter(NelX,NelY,Rmin,x,SenNr); %Apply Filtering Technique
[x]=SERA_Update(NelX,NelY,SenNr,x,Vol,VolObj,i,SR,B,Xmin);
Vol(i)=ceil(sum(sum(x)))/(NelX*NelY);
if i>10;
valoresaltos=sum(Compl(i-4:i)); valoresbajos=sum(Compl(i-9:i-5));
Change=abs((valoresbajos-valoresaltos)/valoresaltos);
end
disp([' Iteration: ' sprintf('%4i',(i-1)) ...
     ' Volume fraction: ' sprintf('%6.3f',Vol(i)) ...
     ' Compliance: ' sprintf('%6.6f',Compl(i))])
figure(1); colormap(gray); imagesc(-x); grid on; axis tight; axis
off; pause(1e-10)
end
disp([num2str(toc),'   RUN FINISHED'])

%------- FE-ANALYSIS -----------------------%
function [U,KE,K]=FE(NelX,NelY,x,Case)
[KE] = lk;
K = sparse(2*(NelX+1)*(NelY+1), 2*(NelX+1)*(NelY+1));
F = sparse(2*(NelY+1)*(NelX+1),1);
for elx = 1:NelX
  for ely = 1:NelY
    n1 = (NelY+1)*(elx-1)+ely;
    n2 = (NelY+1)* elx   +ely;
    edof = [2*n1-1; 2*n1; 2*n2-1; 2*n2; 2*n2+1; 2*n2+2; 2*n1+1;
      2*n1+2];
    K(edof,edof) = K(edof,edof) + x(ely,elx)*KE;
  end
```

```
end
switch Case
    case ('c') % Cantilever
        F(2*(NelX+1)*(NelY+1),1) = -1.0;
        fixeddofs = [1:2*(NelX+1)];
    case ('m') % Beam with a force at the centre of the bottom edge
        F(2*(NelY+1)*(NelX+1),1) = -1/2;
        fixeddofs=union([2*(NelY+1)-1,
          2*(NelY+1)],[2*(NelY+1)*(NelX)+1:2:2*(NelY+1)*(NelX+1)]);
   otherwise
        disp('not assigned to a pre-defined case')
end
alldofs = [1:2*(NelY+1)*(NelX+1)];
freedofs = setdiff(alldofs,fixeddofs);
U = zeros(2*(NelY+1)*(NelX+1),1);
U(freedofs,:) = K(freedofs,freedofs) \ F(freedofs,:);
U(fixeddofs,:)= 0;

%_----------  SERA Update --------------%
function [x]=SERA_Update(NelX,NelY,SenNr,x,Vol,VolObj,i,SR,B,Xmin)
SenNr_min=min(min(SenNr)); SenNr_max=max(max(SenNr));
SenNr_V(1:NelY,1:NelX)=SenNr_min; SenNr_R(1:NelY,1:NelX)=SenNr_max;
for ely=1:NelY
    for elx=1:NelX
        if x(ely,elx)>0.1
            SenNr_R(ely,elx)=SenNr(ely,elx);
        else
            SenNr_V(ely,elx)=SenNr(ely,elx);
        end
    end
end
if Vol(i)>VolObj
    AV(i)=abs(Vol(i)-Vol(i-1));
    AV_Rem=AV(i)*(SR); % Volume to remove in the ith iteration
    NrElem_Rem=max(1,floor(NelX*NelY*AV_Rem))
    [x,NrElem_Rem]=Update_R(NelX,NelY,x,SenNr_R,NrElem_Rem,Xmin);
    if i>2
        NumElem_Add=max(1,floor(NrElem_Rem*(SR-1)));
        [x]=Update_V(NelX,NelY,x,SenNr_V,NumElem_Add);
    end
else
    AV_Rem=B*VolObj;
    NrElem_Rem=max(1,floor(NelX*NelY*AV_Rem));
    [x,NrElem_Rem]=Update_R(NelX,NelY,x,SenNr_R,NrElem_Rem,Xmin);
    NumElem_Add=NrElem_Rem;
    [x]=Update_V(NelX,NelY,x,SenNr_V,NumElem_Add);
end

%-------- UPDATE_VIRTUAL MATERIAL ----------%
function [x]=Update_V(NelX,NelY,x,SenNr_V,NumElem_Add)
SenNr_V_vec=sort(reshape(SenNr_V,(NelX*NelY),1),'descend');
SenNr_V_th=SenNr_V_vec(NumElem_Add,1);
for ely=1:NelY
    for elx=1:NelX
        if SenNr_V(ely,elx)>=SenNr_V_th
            x(ely,elx)=1.;
        end
    end
end
```

```
%-------- SERA_UPDATE_REMOVE ----------------%
function[x,NumElem_Rem]=Update_R(NelX,NelY,x,SenNr_R,NumElem_Rem,Xm
in)
SenNr_R_vec=sort(reshape(SenNr_R,(NelX*NelY),1),'descend');
SenNr_R_th=SenNr_R_vec((NelX*NelY)-NumElem_Rem,1);
NumElem_Rem=0;
for ely=1:NelY
    for elx=1:NelX
        if x(ely,elx)==1
            if SenNr_R(ely,elx)<=SenNr_R_th
                x(ely,elx)=Xmin;
                NumElem_Rem=NumElem_Rem+1;
            end
        end
    end
end

%-------- ELEMENT STIFFNESS MATRIX -----------%
function [KE]=lk
E = 1; nu = 0.3;
k=[ 1/2-nu/6   1/8+nu/8 -1/4-nu/12 -1/8+3*nu/8 ...
   -1/4+nu/12 -1/8-nu/8  nu/6       1/8-3*nu/8];
KE = E/(1-nu^2)*[ k(1) k(2) k(3) k(4) k(5) k(6) k(7) k(8)
                  k(2) k(1) k(8) k(7) k(6) k(5) k(4) k(3)
                  k(3) k(8) k(1) k(6) k(7) k(4) k(5) k(2)
                  k(4) k(7) k(6) k(1) k(8) k(3) k(2) k(5)
                  k(5) k(6) k(7) k(8) k(1) k(2) k(3) k(4)
                  k(6) k(5) k(4) k(3) k(2) k(1) k(8) k(7)
                  k(7) k(4) k(5) k(2) k(3) k(8) k(1) k(6)
                  k(8) k(3) k(2) k(5) k(4) k(7) k(6) k(1)];

%------- MESH-INDEPENDENCY FILTER ----------------%
function [SenNrNew]=Filter(NelX,NelY,Rmin,x,SenNr)
SenNrNew=zeros(NelY,NelX);
for i = 1:NelX
  for j = 1:NelY
    sum=0.0;
    for k = max(i-floor(Rmin),1):min(i+floor(Rmin),NelX)
      for l = max(j-floor(Rmin),1):min(j+floor(Rmin),NelY)
       sum = sum+max(0,Rmin-sqrt((i-k)^2+(j-l)^2));
       SenNrNew(j,i) = SenNrNew(j,i) + max(0,Rmin-sqrt((i-k)^2+
         (j-l)^2))*x(l,k)*SenNr(l,k);
      end
    end
    SenNrNew(j,i) = SenNrNew(j,i)/sum;
  end
end
```

The optimized design of a structure shown in Figure 5 can now be obtained by means of the following function call:

```
SERA_STR(40,40,0.4,0.02,1.2,0.04,1.4,1,0.001,'c')
```

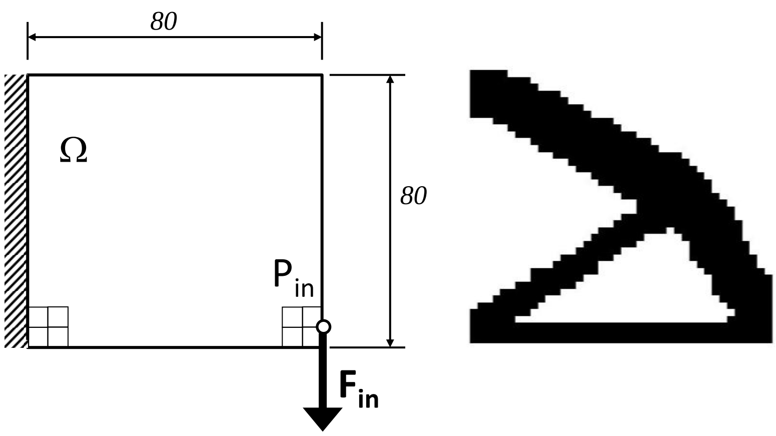

Figure 5: The problem formulation and the optimized design of a clamped beam obtained using the proposed

B. The SERA method for compliant mechanisms topology optimization

The generalized method for structural optimization can be adapted for compliant mechanisms design as follows. Only the general function SERA_COMP changes and the FE function due to the different load cases. The rest of the functions SERA_UPDATE, UPDATE VIRTUAL MATERIAL, UPDATE REAL MATERIAL, lk (Element stiffness matrix) and the FILTER function remains as in the general program and are not repeted here. This means that lines from 55 to 125 remain the same.

The program routine is called with the following command line:

```
SERA_COMP(NelX,NelY,VolObj,PR,SR,B,Rmin,VolIni,Xmin,Case,ksin,ksout)
```

where:
NelX, Number of elements in the X axis (horizontal)
NelY, Number of elements in the Y axis (vertical)
VolObj, Target volume fraction
PR, Progress Ratio
SR, Smoothing Ration
B, Material re-distribution fraction
Rmin, Filtering radius for the filtering technique
VolIni, Initial volume (This code has been simplified to have an initial volume fraction VolIni=1)
Xmin, Minimum density considered, a typical value used is 10^{-4}
Case, Two predefined cases are included: Case=`'i'` for inverter mechanism; and Case= `'m'` for a beam with a force at the centre of the bottom edge

```
%%% SERA method for Structural Top Opt ©CristinaAlonsoGordoa
1    function SERA_COMP(NelX,NelY,VolObj,PR,SR,B,Rmin,VolIni,Xmin,
     Case,ksin,ksout)
2    clc; close all; tic; disp([num2str(toc),'......START.......']);
3    SenNr=sparse(NelY,NelX); x(1:NelY,1:NelX) = VolIni; i = 1;
        Vol(i)=VolIni; Change = 1.; %inicialization of variables
4    while Change >  0.001
5    i = i + 1;
6    Vol(i)= max(Vol(i-1)*(1-PR),VolObj); %For initial design
     domain>VolObj
7    [U,KE,K]=FE(NelX,NelY,x,Case,ksin,ksout);
8    MPE(i)= (U(:,1)'*K*U(:,2)); %MPE in the ith iteration
9    for ely = 1:NelY
10       for elx = 1:NelX
11               n1 = (NelY+1)*(elx-1)+ely;
12               n2 = (NelY+1)* elx   +ely;
13               Ue1 = U([2*n1-1;2*n1; 2*n2-1;2*n2; 2*n2+1;2*n2+2;
                   2*n1+1;2*n1+2],1);
14               Ue2 = U([2*n1-1;2*n1; 2*n2-1;2*n2; 2*n2+1;2*n2+2;
                   2*n1+1;2*n1+2],2);
15               SenNr(ely,elx)=-(Ue1'*KE*Ue2); %Objective: max MPE
```

```
    end
end
[SenNr]=Filter(NelX,NelY,Rmin,x,SenNr); %Apply Filtering Technique
[x]=SERA_Update(NelX,NelY,SenNr,x,Vol,VolObj,i,SR,B,Xmin);
Vol(i)=ceil(sum(sum(x)))/(NelX*NelY);
if i>10;
    valoresaltos=sum(MPE(i-4:i)); valoresbajos=sum(MPE(i-9:i-5));
    Change=abs((valoresbajos-valoresaltos)/valoresaltos);
end
disp([' Iteration: ' sprintf('%4i',(i-1)) ...
     ' Volume fraction: ' sprintf('%6.3f',Vol(i)) ...
     ' Compliance: ' sprintf('%6.6f',MPE(i))])
figure(1); colormap(gray); imagesc(-x); grid on; axis tight; axis
off; pause(1e-10)
end
disp([num2str(toc),'   RUN FINISHED'])

%------- FE-ANALYSIS -----------------------%
function [U,KE,K]=FE(NelX,NelY,x,Case,ksin,ksout)
[KE] = lk;
K = sparse(2*(NelX+1)*(NelY+1), 2*(NelX+1)*(NelY+1));
F = sparse(2*(NelY+1)*(NelX+1),2); U =
sparse(2*(NelY+1)*(NelX+1),2);
for elx = 1:NelX
  for ely = 1:NelY
    n1 = (NelY+1)*(elx-1)+ely;
    n2 = (NelY+1)* elx   +ely;
    edof = [2*n1-1; 2*n1; 2*n2-1; 2*n2; 2*n2+1; 2*n2+2; 2*n1+1;
       2*n1+2];
    K(edof,edof) = K(edof,edof) + x(ely,elx)*KE;
  end
end
switch Case
    case ('i')% DISPLACEMENT INVERTER
        din=(NelY+1);
        dout=2*NelX*(NelY+1)+(NelY+1);
        K(din,din)=K(din,din)+ksin;
        K(dout,dout)=K(dout,dout)+ksout;
        F(din,1)=-1;
        F(dout,2)=1;
        fixeddofs=union([1:1:6],[(2*(NelY+1)-6):1:2*(NelY+1)]);
    case ('c')% CRUNCHING MECHANISM
        din=(NelY+1);
        dout1=2*NelX*(NelY+1)+2;
        dout2=2*(NelX+1)*(NelY+1);
        F(din,1)=-1;
        F(dout1,2)=1;
        F(dout2,2)=-1;
        K(din,din)=K(din,din)+ksin;
        K(dout1,dout1)=K(dout1,dout1)+ksout;
        K(dout2,dout2)=K(dout2,dout2)+ksout;
        fixeddofs=union([1:1:2*(ceil(0.01*NelY)+1)],[(2*(NelY+1)-
          (2*ceil(0.01*NelY)+1)):1:2*(NelY+1)]);
    otherwise
        disp('case not pre-defined')
end
alldofs = [1:2*(NelY+1)*(NelX+1)];
freedofs = setdiff(alldofs,fixeddofs);
U(freedofs,1) = K(freedofs,freedofs) \ F(freedofs,1);
U(freedofs,2) = K(freedofs,freedofs) \ F(freedofs,2);
U(fixeddofs,:)= 0;
```

The optimized design shown in Figure 6 can now be obtained by means of the following function call:

```
SERA_COMP(40,40,0.4,0.02,1.4,0.004,1.2,1,0.001,'i',1,1)
```

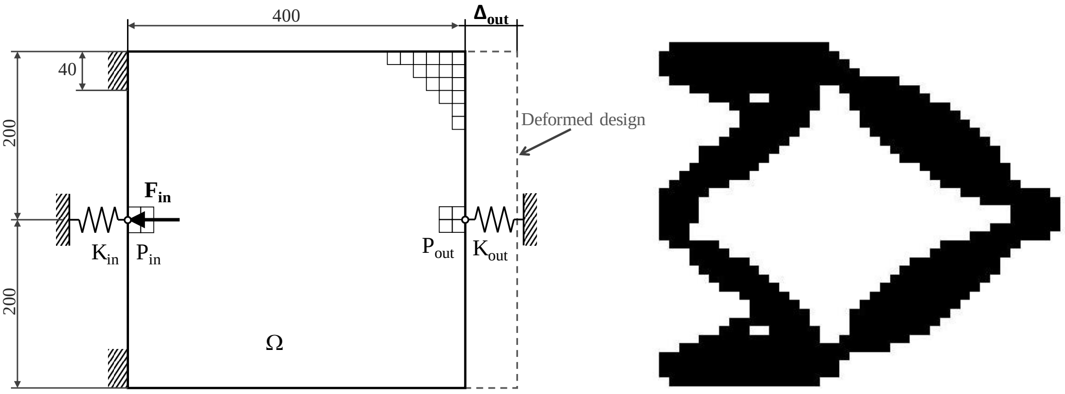

Figure 6: The problem formulation and the optimized design of a compliant mechanism obtained using the proposed

IV. Conclusion

A simple Matlab code to design structures and compliant mechanisms with the Sequential Element Rejection and Admission method has been presented in this paper for demonstration and educational purposes.

The SERA method allows material to be added and removed from the design domain until the optimum topology is reached. The main difference of this method with respect to other bi-directional methods that add and remove elements from the design domain is the separate treatment of ‘real’ and ‘virtual’ material. Separate criteria for each material model are defined to efficiently add and remove elements and achieve the optimum topology.

Two benchmark problems are used as examples of function calls to the Matlab code. The two examples, as well as multiple examples presented by the authors in previous papers serve as demonstration of the validity of the proposed method to design both structures and compliant mechanisms by means of topology optimization techniques.

Acknowledgments

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Iso-contour method for optimization of steered-fiber composites

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I. Introduction

Advanced composite materials are used more and more in industrial applications for the aircraft and aerospace structures. Due to their superior and flexible mechanical properties and low weight the composite materials can be an attractive alternative to metals. Especially for the applications when assembly simplification, high strength, high stiffness and good fatigue resistance are of interest.

One of commonly used composites are fiber-reinforced laminates. All fibers are usually aligned in parallel in each layer. The limitations of this approach can be overcome by more complex fiber alignments, when fiber directions can be tuned to improve specific load-carrying capabilities of the composite part. For example, to reduce stress concentration of a composite plate with holes, curved-fiber composites can be used. Several different methods were proposed to find optimal fiber orientation distributions. Gürdal and Olmedo [1] used angle variations for continuous linear fiber. They introduced a fiber path definition and formulated closed-form and numerical solutions for simple rectangular plates. Hyer and Charette [2] proposed to choose the fiber orientations so that the fibers in a particular layer were aligned with the principal stress directions in that layer. Hansel and Becker [3] proposed a heuristic optimization algorithm for minimum weight design of composite laminates based on layer-wise removal of elements with low stress measures.

IJsselmuiden et al. [4] used the fiber angles at the nodes of an FE model as design variables. Their approach is adapted to use lamination parameters.

Each of these approaches has its drawbacks and benefits. For example, approaches, working with predefined path types cannot allow arbitrary fiber paths. Methods, based on node/element variation of fiber orientation usually end up with a large number of design variables, which makes it challenging to find the optimal design. Also, obtained optimal solutions are often non-manufacturable due to non-smooth fiber paths.

The research presented in this paper is focused at solving the described difficulties by allowing more flexibility to the fiber-paths definition and by creating optimal smooth manufacturable fiber paths; at the same time the number of design variables is kept relatively low. All these requirements should ideally result in a powerful method for the fiber-steered composite optimization for shell structures.

In this paper basic ideas of the method for the 2D fiber steering optimization are presented. The main idea is somehow related to the level-set method: in order to steer the fibers, iso-contour lines of an artificial hyper-surface, defined over 2D geometry domain, are used. Thus, the smoothness of this artificial surface can guarantee continuity/smoothness of the obtained fiber paths. This can be used to create manufacturable steered fiber composites. Finally, by modifying this artificial surface, we can control the fiber paths and optimize the design of a composite part for the specific needs.

II. Method

In this paragraph, the main steps of the proposed method are described. As was mentioned before, the artificial surface should be defined over the 2D domain in order to control fiber paths. Overlaid control points are defined for 2D geometry domain similar with mesh, which will be used to control the artificial surface. Number and location of the points can be varied. Then a “height” at each of these control points is defined and an interpolating artificial surface is fitted over all control points. In the current work a non-parametric surface fit, based on Kriging is used, which allows also non-grid positions of the control points. Other types of the fitting surfaces will be studied later. In analogy to a geodesic map, the iso-contour lines can be obtained from this artificial interpolating surface, which will steer the fiber paths at each point of the 2D geometry domain. Finally, the “heights”, defined in the control points are the design variables, which means, that the numbed of control points represents the dimension of the optimization problem. At the same time, the distribution of control points can be adjusted to fit specific geometry elements (e.g. holes, etc.) and to allow more flexibility for fiber paths in some regions. In the current work, a simple evolutionary algorithm is used to find the optimal fiber paths orientations.

The proposed method was implemented in python class. For solving the FE problem, the ANSYS software is used. DAKOTA software is used for the optimization. The fiber direction for each finite element of the FE

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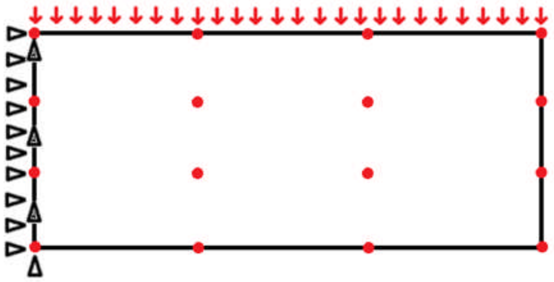

Figure 1. The plate with the uniformly distributed loading and overlaid control points

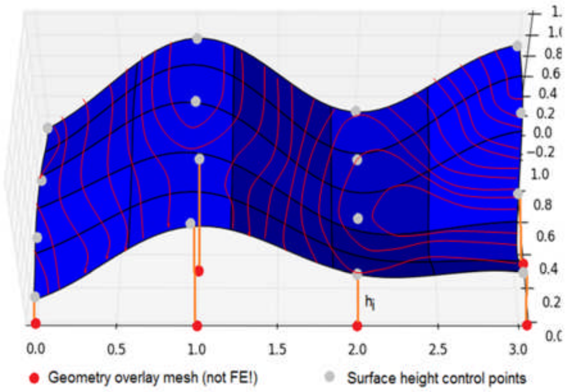

Figure 2. Interpolating 3D surface with iso-lines

model is defined as the artificial surface gradient vector (the normal to the iso-contour line) at the finite-element geometrical center.

III. Simple application example

As first example to test the method, a simple plate in a bending problem with uniformly distributed top in-plane loading is considered as shown in Fig. 1. Dimensions of the plate are defined similar to Setoodech [5], with the aspect ratio of the sides equal to three and a uniformly distributed load. The left side of the plate is fixed.

For the FE analysis 2nd order shell elements (SHELL281) are taken. These elements can be used for modeling layered composite shells or sandwich constructions. The number of design variables is equal to 16 (4×4) control points of the “mesh” Fig. 1.

The optimal result was obtained after 2040 iterations. Fig. 2 shows the optimal artificial 3D surface configuration with iso-lines, from which optimal fiber directions are obtained. The fiber paths will be perpendicular to the surface iso-lines Fig. 3.

As can be seen in Fig. 3, we obtained reasonable result in comparable to those given in [5]. On the other-hand one potential difficulty of the method was identified. Because of the fact that for a given iso-contour lines multiple corresponding 3D surfaces can be created, it can be difficult for the optimizer to find the global best solution. Additional research is needed to find how to solve that problem.

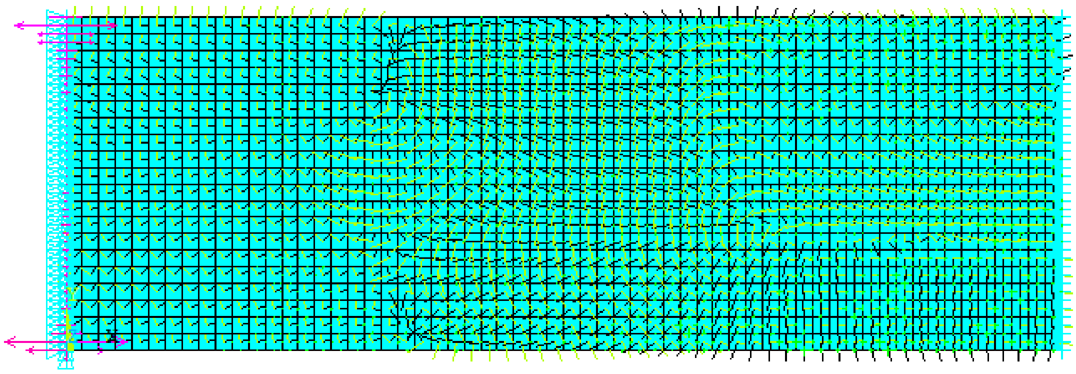

Figure 3. Comparison of the fiber angle at each element.

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Design Space Exploration Based on the Analysis of the Inner Structure of Sensitivities

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This contribution is concerned with the analysis of the internal structure of sensitivities of engineering structures with respect to modifications in shape. Solving a given structural optimization problem is almost an automatic process, where the sensitivities are only computed to serve the mathematical optimizer. However, the process of definition of a structural optimization problem is human controlled and is based on his/her experience and knowledge. Within this process decisions which tackle the kind of design parametrization, the type and number of design variables and relations between these variables have an extraordinary impact on the quality of optimization results, on the solubility of the problem and on the corresponding computational effort. The exploration of structural design is utilized to facilitate and substantiate these decisions.

Such exploration is usually based on parameter studies, which are evaluated using standard statistical methods. This contribution outlines an enhanced analysis of the design sensitivities beyond the standard computation of the gradient values. It is based on the analytical derivation and efficient computation of the Fréchet derivatives of objectives and constraints with respect to the full space of all possible design variables. This overhead of sensitivity information is examined by a singular value decomposition (SVD) and principal component analysis (PCA) in order to detect major and minor influence of model parameters on the structural response, the objectives and constraints. Thus, this methodology leads to valuable qualitative and quantitative insight, which is so far unused in standard approaches to structural optimization. This knowledge enables the optimizer to understand and improve the models systematically.

The generic concept is applied to shape optimization of shell structures which are modeled by nonlinear solid shell elements proposed by Klinkel et al. [1,2]. The design of such structures is extremely important for their stability, robustness and load-bearing capacity. The variational design sensitivity analysis for this nonlinear solid shell is performed and especially the pseudo load matrix and the sensitivity matrix are derived by Gerzen et al. [3]. Illustrative examples demonstrate the advocated concept.

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Multidisciplinary Optimization of a Transonic Fan Blade for High Bypass Ratio Turbofan Engines

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This paper presents the multidisciplinary and multiobjective optimization of a transonic fan blade for a high bypass ratio turbofan engine. Aerodynamic as well as static and dynamic structural performance criteria are considered in the optimization process. A two level optimization strategy is applied consisting of a Differential Evolution algorithm coupled to a Kriging surrogate model and high-fidelity Computational Fluid Dynamics and Computational Structural Mechanics analysis tools. The fan blade is designed for a long-range aircraft mission. The first objective in the optimization is therefore the maximization of the efficiency at cruise conditions. A second objective is defined in order to keep the vibration response of the fan as low as possible within its operating range. In addition several aerodynamic and structural constraints are imposed. An efficiency gain at the design point of 2.47% and an overall improvement of the vibration response of the fan blade prove the effectiveness of the optimization system.

Nomenclature

CFD	=	Computational Fluid Dynamics
CSM	=	Computational Structural Mechanics
DE	=	Differential Evolution
FEM	=	Finite Element Method
RANS	=	Reynolds-averaged Navier-Stokes

I. Introduction

High bypass ratio turbofan engines are nowadays the de-facto standard for powering medium and long range aircraft due to their high thrust and good fuel efficiency up to high subsonic aircraft speeds. One of the central components of the turbofan engine is the fan, which generates the majority of the engine's thrust and plays a key role for its fuel efficiency. Besides the apparent need to be aerodynamically efficient in order to reduce engine fuel consumption, the fan blades need to withstand considerable static and dynamic structural loads to which they are subjected to during operation. The design process of fan blades is therefore a multidisciplinary problem. Further complexity is added to the design problem by a high level of interaction between the different disciplines, which prevents one discipline to be optimized in isolation if a global optimal solution is sought. In this paper a multidisciplinary and multiobjective optimization system is presented and applied to the design of a transonic fan blade for a high bypass ratio turbofan engine. Structural and aerodynamic performances are treated concurrently in the optimization process, therefore allowing to find global optimal solutions in a limited design time.

II. Optimization System

The optimization system shown in Fig. 1 is the result of more than one and a half decades of research and development at the von Karman Institute[1,2]. Its core components are a multi-objective Differential Evolution algorithm [3], a database, several metamodels, including Radial Basis Functions, Artificial Neural Networks and Kriging, and a high-fidelity evaluation chain including a fully automatic geometry and CAD generation, automatic meshing and high-fidelity performance evaluations by Computational Fluid Dynamics (CFD) and Computational Structural Mechanics (CSM). The optimization system is based on a two-level approach with a Kriging metamodel being applied in the present application. An initial sampling of the design space is

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performed using a fractional factorial design containing 65 samples, whereas each sample is analyzed by the high-fidelity evaluation chain. The resulting relation between design parameters and performance is stored in the database which is used to generate a Kriging metamodel. Subsequently, the Differential Evolution algorithm is applied to find the best designs based on the metamodel predictions. A number of these designs are then re-evaluated by the high-fidelity evaluation chain and the results are added to the database, which is used to re-generate the metamodel. This process is expected to increase the prediction accuracy of the metamodel in the regions where it previously predicted optimal designs. In this paper, an entire loop consisting of metamodel generation, DE optimization, high-fidelity re-evaluation and storage of the results in the database is termed *iteration*.

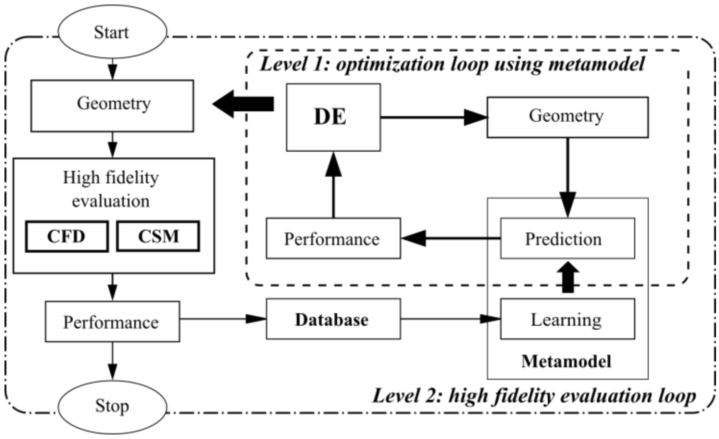

Figure 1. Flow chart of the optimization system.

III. Fan Blade Parametrization

The geometry of the fan blade is defined by parametric Bézier and B-Spline curves which specify the blade chord, blade angles, the thickness distributions at hub and tip sections and the profile stacking axis by lean and sweep. The control points of the parametric curves are used as optimization parameters. A total of 31 optimization parameters are defined in this work, allowing all of the above mentioned quantities to change within specified limits during the optimization process.

IV. High-Fidelity Performance Evaluations

The commercial 3D Reynolds-averaged Navier-Stokes solver FINE™/Turbo is used for the aerodynamic performance evaluations of the fan blade. The fluid-domain is discretized with a multi-block structured mesh consisting of 1.8 million grid points. Five operating points on a constant speedline are computed by varying the outlet pressure. Turbulence effects are computed with the one-equation Spalart-Allmaras turbulence model.

The solid domain of the fan is discretized with an unstructured mesh consisting of quadratic tetrahedral elements. The open-source Finite Element Solver CalculiX[4] is used for the structural analyses which consist of static stress and vibration analyses. The static stresses are computed using non-linear geometric analyses. The blade is subjected to centrifugal loads at take-off conditions and pressure loads, which are obtained from the CFD computations and interpolated onto the FEM grid. The vibration of the fan blade is assessed by modal analyses whereas centrifugal stiffening of the structure is included in the computations. Blade vibrations are computed at three main operating points; take-off, top of climb and cruise. The Campbell diagram is used to compute the margin between excitation frequencies and blade natural frequencies at the rotational speeds associated with the aforementioned operating points. Excitations from one per revolution and two per revolution disturbances are considered covering possible sources like unbalance and cross-wind. Resonance is computed for the first ten harmonics of each excitation source and the first four Eigenmodes of the fan blade. The blade is modeled using material properties of Titanium.

V. Objectives and Constraints

Two objectives and five constraints are specified in this optimization. The first objective is the maximization of the isentropic total-to-total efficiency of the fan blade at the design point mass flow of 576 kg/s at cruise conditions. The second objective is the maximization of the margins between excitation and natural frequency of the fan blade for the three critical operating conditions take-off, top of climb and cruise.

As a first constraint, the minimal stall margin is specified. This margin specifies the mass flow difference between the design point and the point where the flow through the fan becomes unstable and stall or surge occurs. The second constraint is the design requirement that the total pressure ratio at the design point needs to be equal or bigger than 1.5. The third and fourth constraints are defined to ensure that the design point mass flow of 576kg/s is within the stable operating range of the fan. More specifically, the stall point (i.e. the last stable operating point towards low mass flows) needs to have a mass flow which is lower than the design point mass flow. Consequently, the choke point, which is the operating point with the highest mass flow in the fan's operating range, needs to be at a higher mass flow than the specified design mass flow. The fifth and last constraint is defined to ensure that the maximum von Mises stresses in the fan blade are lower than the yield strength of the titanium material.

VI. Results

Figure 2 shows the objective space after a total of 16 iterations. All symbols in the figure indicate a design which is satisfying all of the above specified constraints. The orange diamond indicates the baseline design. It should be noted that the maximization problem was redefined to a minimization problem; therefore improved performance is obtained towards the lower left corner of the objective space. As can be observed, a considerable performance gain of 2.47% in efficiency and 4.83 in frequency margin were obtained with respect to the baseline design after only five optimization iterations, which is equal to overall 95 high-fidelity performance evaluations. It is quite a remarkable result that the continuously updated Kriging metamodel is able to guide the optimization towards feasible regions in the design space and enables to find designs with improved performance within only a very limited number of iterations, although the optimization problem includes quite restrictive constraints. In total only 32 out of 161 designs that have been evaluated by the high-fidelity tools are actually satisfying all of the imposed constraints.

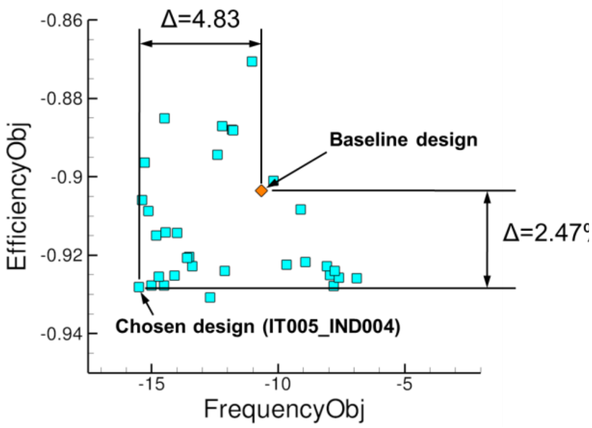

Figure 2. Objective space after 16 iterations. Only designs which are satisfying the constraints are shown.

As shown on the left hand side of Fig. 3, the optimized design shows a considerable efficiency improvement over the entire operating range. It is noticeable that the higher efficiency does not come at the cost of a narrower operating range compared to the baseline design. Solely a slight shift of the entire range towards higher mass flows is obtained. The pressure ratio has been slightly increased as well as shown on the right hand side of Fig. 3 and is well above the imposed constraint value of 1.5 (as indicated by the dashed line).

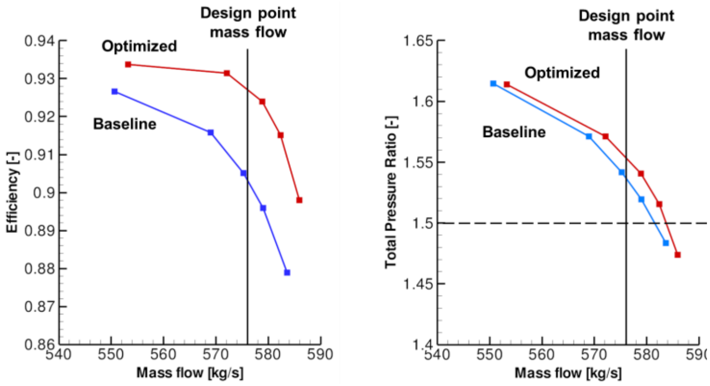

Figure 3. Performance curves of the baseline and the optimized designs.

Acknowledgments

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Optimization of Curved Fibre Composites for Buckling Performance with Manufacturing Constraints

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I. Minimising mass of a composite aerospace wing panel

WING panels are large, thin plates subjected to tensile and compressive loads during flight. Such panels are designed for minimum mass, thereby maximising fuel efficiency and a capacity to carry loads sufficient to prevent buckling or material failure. Panels are typically manufactured from laminated carbon fibre reinforced plastic (CFRP) where straight fibres are supported by a resin matrix. Design using CFRP is dominated by highly orthotropic material properties and complex failure modes. By correlating the material orthotropy with structural and loading anisotropy a mass reduction can be achieved. One method for tailoring structural properties to achieve lower mass is to curve fibres during manufacture such that the inherent orthotropy in the composite material provides optimal structural properties.

There are two manufacturing methods available for curved CFRP, both based on depositing layers of tape along a path. Advanced Fibre Placement (AFP) can follow gently curving paths without a change in layer thickness. Continuous Tow Shearing (CTS) allows tighter bend radii and layer thickness changes as a function of fibre angle.

In each method fibre curvature is constrained by manufacturing capability and a compromise is struck between maximum fibre curvature and deposition rate. The CTS method is currently limited by deposition rate to demonstration panels and prototypes. The research presented here is an investigation into the relationship between mass and fibre curvature, with the aim of justifying the investment necessary to scale CTS to industrial production rates. AFP is a current industrial process capable of rapid deposition but is limited in the curvature of fibres that can be achieved without significant defects. The method allows the potential each method has for mass reduction to be compared.

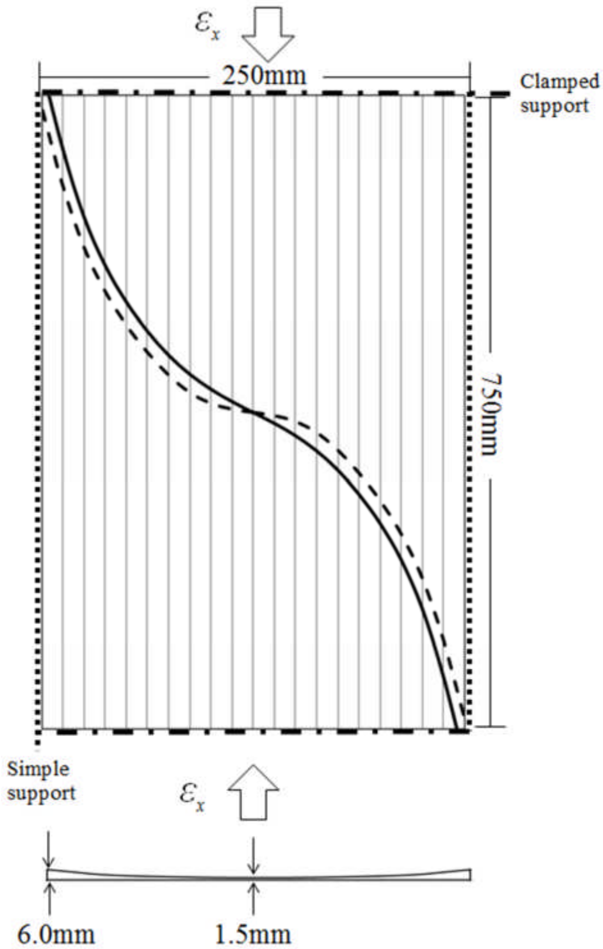

Figure 1. Example curved fibre panel, not to scale
The vertical lines denote the discretisation strips used to evaluate the buckling performance of the panel. The curved fibre path gives rise to the variation in thickness indicated in the lower image when using the CTS manufacturing technique.

II. Summary of the optimisation problem

The objective is to minimise panel mass subject to a set of constraints. The problem complexity lies in the manufacturing and buckling constraints.

The buckling analysis used in this paper is the VIPASA[1] infinite strip method. This restricts the design space to prismatic panels in exchange for significantly reduced calculation time relative to commercial Finite Element Analysis.

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There are nonlinear constraints on the allowable stress and strain within the panel and on the critical panel buckling load. Critical buckling can switch from one mode to another, particularly in the vicinity of an optimum design. There are an integer number of layers in the structure; the nominal thickness of each is limited to stock material sizes. When considering the CTS technique, in which dry fibres are sheared before impregnation with resin film[2], layer thickness and fibre angle variables are sinusoidally coupled, see Fig. 1.

The manufacturing constraints are incorporated by mapping a vector of control variables onto a parameterised curve. The entire design space therefore comprises designs which meet the curvature constraint, at the cost of introducing a non-linear mapping between design variables and the resulting design. Changing any one of these control variables changes the fibre angle throughout the entire structure. Changing fibre angle changes thickness and other structural properties. Consequently the response to varying this control vector is complicated. The design problem is non-linear, multi-modal, partially discrete, discontinuous and tightly coupled. This constitutes a challenging applied optimisation problem.

III. Preliminary results and discussion

The first optimisation method used was a conjugate gradient solver with multiple initial designs. This fails to resolve plateaus in the design space, is readily trapped in local minima and resolves coupled variables poorly. Differential evolution (Latin hypercube initialisation, random algorithm parameters) resolved the limitations of gradient search, although the method converges prematurely, it consistently outperforms a pure gradient search in terms of the quality of the optimum found. Since this genetic algorithm requires a very large number of function evaluations to make progress, it is slow with simple designs and infeasible with complex designs. It has nevertheless been sufficient to map the design space for a simple panel.

Figure 2 shows the effect of allowable radius of fibre curvature on panel mass, converging asymptotically on the straight fibre case at large radii. The data points show optima found using differential evolution from unique initial populations. Clearly, the highest mass reduction for CTS optima, approximately 40% relative to straight fibre, would be a very good result to reproduce in physical testing.

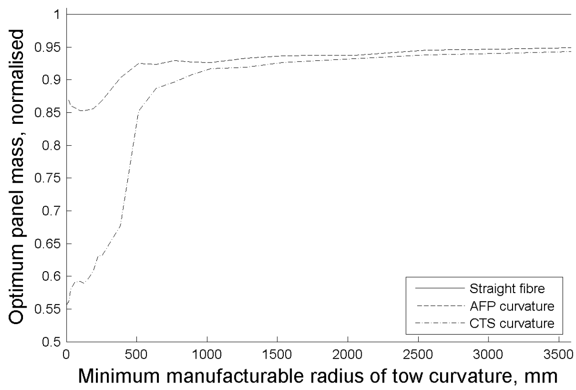

Figure 2. Calculated mass reduction vs radius of tow curvature

The mass is normalised by the straight fibre case. Using CTS, which couples thickness and fibre angle, allows a 40% reduction in mass. It should be noted that current manufacturing methods can achieve approximately 600mm radius with AFP and 30mm with CTS.

IV. Conclusions and ongoing work

Differential evolution is sufficient to determine the general properties of the design space. It is simple to implement, robust and reliable. However, repeated runs of the calculation converge to different local optima. This provides evidence that the global optimum is not yet reliably identified. Nevertheless, the preliminary mass reduction identified using the genetic algorithm is good enough to justify further development efforts.

The current genetic algorithm will be replaced with a hybrid optimisation scheme based on regression Kriging. This provides a means of estimating the probability that the local optima found approximates a global optima as well as an elegant way to incorporate hierarchical or multi-level methods. This will facilitate study of more complicated design problems, in the sense of greater numbers of design variables and integrating stiffeners with the panel geometry.

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The 10th ASMO UK / ISSMO Conference on Engineering Design Optimization Product and Process Development

Metamodelling and Optimisation in Two-Scale Elastohydrodynamic Lubrication

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Abstract

Two-Scale Elastohydrodynamic Lubrication (EHL) is a novel method for investigating the role of topography in contacts such as mechanical bearings. Based on the Heterogeneous Multiscale Methods (HMM) [1] two separate scales are defined, the large scale describes the bearing domain and the small scale describes the Fluid Structure Interaction (FSI) of topographical features. Small scale simulations are treated as near-periodic, meaning that they represent a single point at the large scale. The solutions at both scales are coupled via a pressure gradient – mass flow rate relationship which is used in place of the conventional lubrication approximation in the bearing domain [2]. A Design of Experiments (DoE) of small scale simulations is created using an Optimal Latin Hypercube (OLHC) to cover the range of variables needed by the large scale solution procedure. The small scale solutions are subsequently represented at the large scale through homogenisation and Moving Least Squares (MLS) metamodels [3]. These metamodels are validated through Cross Validation (CV) techniques such as k-fold CV and Leave-One-Out CV [4]. Topography is parameterised at the small scale such that through the use of MLS metamodels optimisation of these features is achieved at the large scale. Optimisation at the large scale is measured through the minimal coefficient of friction achieved at constant load when a constant topography is applied over the length of bearing.

Keywords: EHL; Metamodelling; Optimization

Graphical Summary

A graphical summary of the main components of the two-scale method for EHL is presented in *Fig. 1*. The two-scale method does not form an iterative cycle, *Fig. 1* highlights the flow of information from one component to the next and the solution processes which occur at each stage. Dotted arrowed lines indicate that all information required must be passed as a prerequisite to the next stage, a solid arrowed line is used where information is passed during the solution process, and cyclical arrowed lines represent where FSI occurs at each scale.

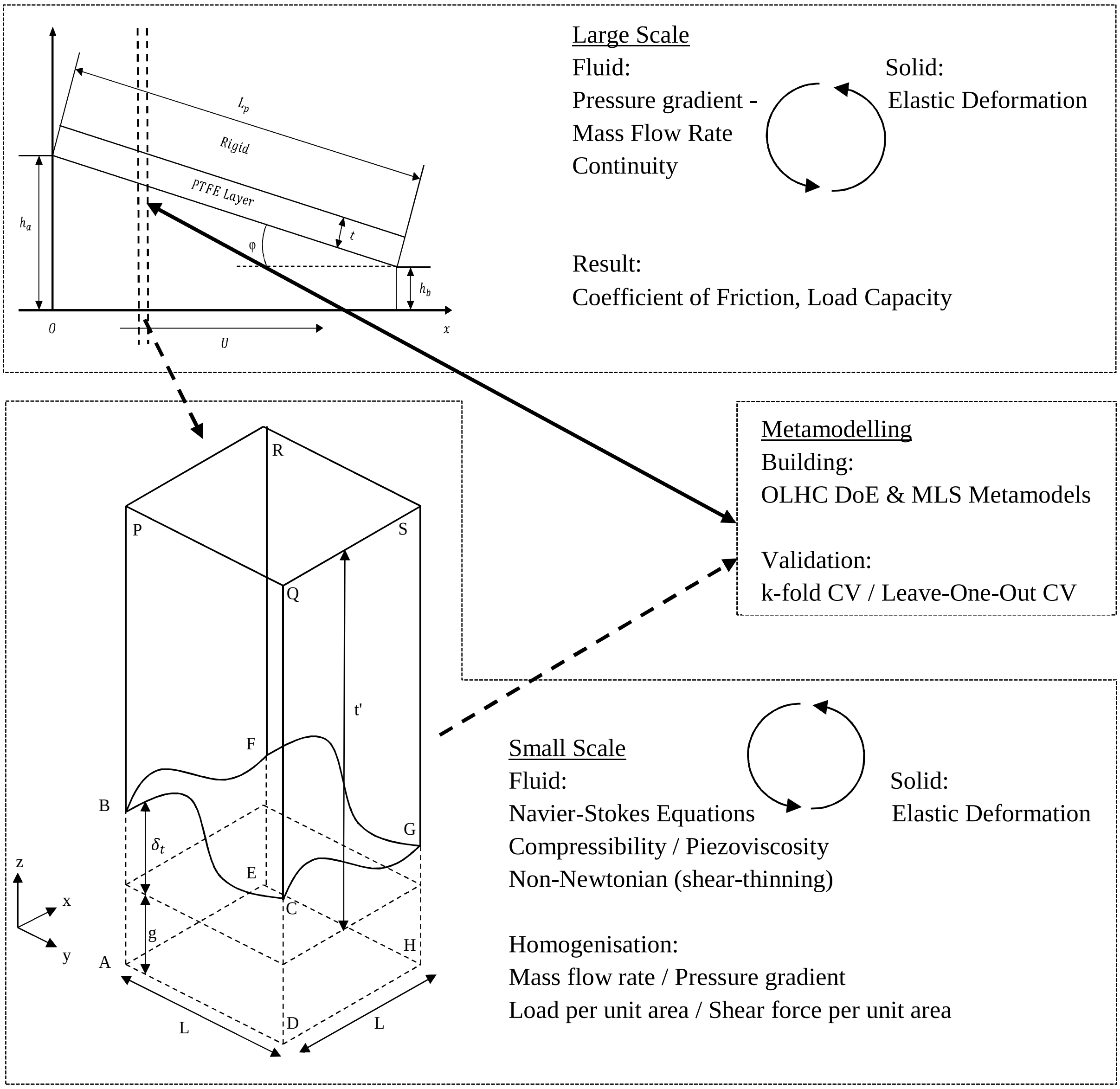

Fig. 1 – Graphical summary of the two-scale method for EHL

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MULTI-OBJECTIVE OPTIMIZATION of REAL-LIFE COMPLEX OBJECTS USING IOSO TECHNOLOGY

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ABSTRACT

This paper presents the main capabilities of IOSO (Indirect Optimization based on Self-Organization) technology algorithms, tools and software, which can be used for the optimization of complex systems and objects. IOSO implements a novel evolutionary response surface strategy. This strategy differs significantly from both the traditional approaches of nonlinear programming and the traditional response surface methodology. That is why, IOSO algorithms have higher efficiency, provide a wider range of capabilities, and are practically insensitive with respect to the types of objective function and constraints. They could be smooth, non-differentiable, and stochastic, with multiple extrema, with the portions of the design space where objective function and constraints could not be evaluated at all, with the objective function and constraints dependent on mixed variables, etc. The capabilities of IOSO software are demonstrated using well-known test problems of solving complex single-objective and multi-objective problems.

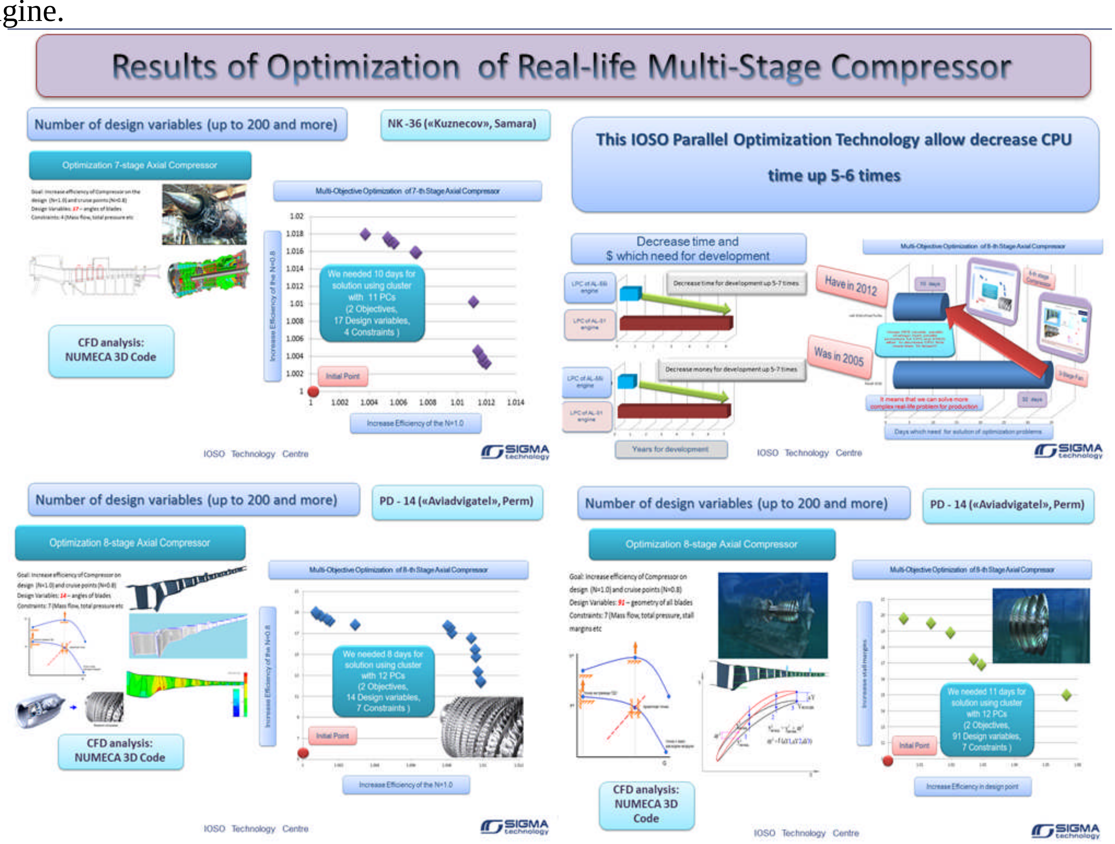
Our approach is based on the widespread application of the response surface technique, which depends upon the original approximation concept, within the frameworks which adaptively use global and middle-range multi-point approximation. One of the advantages of the proposed approach is the possibility of ensuring good approximating capabilities using the minimum amount of available information. This possibility is based on self-organization and evolutionary modeling concepts. During the approximation, the approximation function structure is being evolutionarily changed, so that it allows for the successful approximation of the optimized functions and constraints having sufficiently complicated topology. The obtained approximation functions can be used in multi-level procedures with the adaptive change of simulation levels within both single and multiple disciplines of object analysis, and also for the solution of their interaction problems.

Every iteration of IOSO consists of two steps. The first step is the creation of an analytical approximation of the objective function(s). Each iteration represents a decomposition of the initial approximation function into a set of simple approximation functions. The final response function is a multi-level graph. The second step is the optimization of this approximation function. This approach allows for corrective updates of the structure and the parameters of

the response surface approximation. The distinctive feature of this approach is an extremely low number of trial points to initialize the algorithm. The obtained response functions are used in the multi-level optimization while adaptively utilizing various single and multiple discipline analysis tools that differ in their level of sophistication. The optimization of the response function is performed only within the current search area during each iteration of IOSO.

This step is followed by a direct call to the mathematical analysis model or an actual experimental evaluation for the obtained point. During the IOSO operation, the information concerning the behavior of the objective function in the vicinity of the extremum is stored, and the response function is made more accurate only for this search area. While proceeding from one iteration to the next, the following steps are carried out: modification of the experiment plan; adaptive selection of the current extremum search area; choice of the response function type (global or middle-range); transformation of the response function; modification of both parameters and structure of the optimization algorithms; and, if necessary, selection of new promising points within the researched area. Thus, a series of approximation functions for a particular objective of optimization is built during each iteration. These functions differ from each other according to both structure and definition range. The subsequent optimization of these approximation functions allows us to determine a set of vectors of optimized variables.

This paper demonstrates the optimization of parameters for multi-stage axial real-life compressor using 3-D code of NUMECA. Some of these results were realization for modern air engine.



Reliability Based Optimization of Composite Structures Based on Most Probable Point (MPP)

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This paper focuses upon the efficient reliability based optimization of composite structures. There are several reliability analysis methods including Monte Carlo Simulation method, Importance Sampling method and MPP-based methods. The main advantage of using MPP-based methods is that they are relatively computationally more efficient. Performing reliability-based optimization needs a double loop process. In the inner loop probability of failure and therefore reliability index is obtained then the outer loop would search for the optimum design point. Here, the MPP-based method is utilized in the inner loop for reliability assessment which itself is an optimization process for finding MPP. First, a short column is used to verify the results of method. Then this method is applied to optimization of laminated composite structures. The results obtained using MPP-based reliability analysis of a composite structure are compared with Monte Carlo Simulation results. It is shown that MPP-based methods performed better than Monte Carlo Simulation in terms of speed, however, Monte Carlo Simulation method is more accurate and robust than MPP-based methods.

Nomenclature

MPP	=	Most Probable failure Point
σ	=	Standard Deviation of the random variable
μ	=	Mean Value of the random variable
$p_x(X)$	=	Probability Density Function of random variables
$g(X)$	=	limit state function
β	=	Reliability Index

I. Introduction

One of the main challenges in engineering design is uncertainty. The majority of the structural design parameters have random nature. Uncertainties include changes in parameters such as material properties. This uncertainty may lead to instability and finally structural failure. Therefore uncertainty should be considered in design.

This paper presents a method in which search for the most probable failure point is performed. During the search the failure surface is approximated by a geometric form. Actually MPP is a point on the limit state function which is the nearest point to the origin of the standard normal space. The distance between the origin and MPP is referred to as the reliability index as shown in Figure 1. This point is known as the point that has the highest amount of probability density function. There are two approaches to find MPP that are Reliability Index Approach (RIA) and Performance Measure Approach (PMA). In RIA the algorithm is to search for the shortest distance between failure surface $g(X)=0$ and origin of the standard normal space that is the reliability index,

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and in PMA the reliability index is fixed thus the algorithm searches for minimum of limit state function. According to the problem requirements an appropriate algorithm would be chosen.

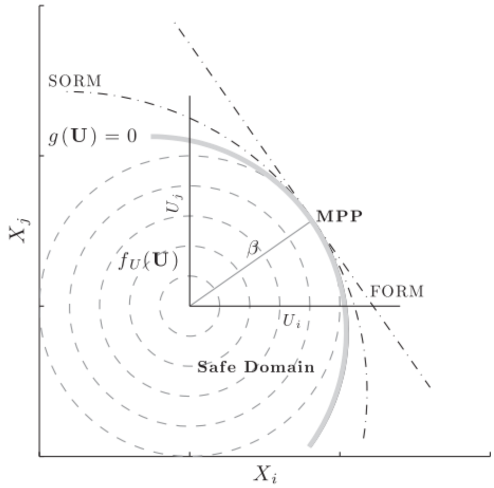

Figure 1. Schematic representation of FORM/SORM approximations [1]

II. Preliminary Results

The method is applied to an isotropic short column for verification of results. Preliminary results are compared to those of Dubourg [2] (Table 2). Following the verification, the methodology is applied to laminated composite structures. Geometry of the composite structure is shown in Figure 1.

- Short Column

This test problem involves the plastic analysis and design of a short column with rectangular cross section (width b and depth h) having uncertain material properties (yield stress) and subject to uncertain loads (two bending moments M1 and M2 whose axes are defined with respect to the two principal axes of inertia of the cross section as well as axial force F). The performance function, designed to represent the stress in the column at which the yield stress is exceeded, is defined as:

$$g = 1 - \frac{4M_1}{bh^2\sigma_y} - \frac{4M_2}{b^2h\sigma_y} - \left(\frac{F}{bh\sigma_y}\right)^2$$

The distributions for F, M_1, M_2, and σ_y are presented in Table 1. Failure for this performance function is defined by $g \le 0$. An objective function of cross-sectional area and a target probability of failure are used in the design problem:

$$\begin{aligned}
\min \quad c(d) &= c_0(d) + p_f(d)c_f(d) \\
&= c_0(d)\left(1+100p_f(d)\right) \\
&= \mu_b\mu_h\left(1+100p_f(d)\right)
\end{aligned}$$

$$s.t. \quad \beta^* \ge 3$$

$$100 \le \mu_b, \mu_h \le 1000$$

$$1/2 \le \frac{\mu_b}{\mu_h} \le 2$$

Where $p_f c_f$ is the expected failure cost which is chosen here to be proportional to the construction cost c_0 . The geometrical constraints are mentioned in the formulation.

Table 2 gives a summary of the results from MPP-based method.

Table 1. Statistical properties of random variables

Variable	**Distribution**	**Mean**	**C.o.V.**
M_1 (N.mm)	Lognormal	250×10^6	**30%**
M_2 (N.mm)	Lognormal	125×10^6	**30%**
P (N)	Lognormal	2.5×10^6	**20%**
σ_y (MPa)	Lognormal	40	**10%**
b (mm)	Normal	μ_b	**1%**
h (mm)	Normal	μ_h	**1%**

Table 2. Results for the short column under oblique bending

	Opt. design	Cost Function (mm^2)	Reliability Index
MPP-based RBDO	b=399 h=513	2.12×10^5	3.38

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The Curse of Numerical Noise and Implications for CFD-Based Design Optimization

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Numerical noise is an inevitable by-product of Computational Fluid Dynamics (CFD) simulations which, in the context of design optimization, can lead to challenges in finding optimum designs. A number of factors serve to influence the level of noise present: these include the choice of turbulence model and the size, type and density of cells in the computational grid. This article draws attention to numerical noise illustrating the difficulties it can cause for road vehicle aerodynamics simulations.

Firstly a benchmark problem, flow past the Ahmed body, is used to assess a range of turbulence models and grid types. The Ahmed body is suitable for this purpose because the surrounding flow field retains the salient features of bluff-body vehicle aerodynamic flows, yet it is simple to model from a geometrical viewpoint as illustrated in Figure 1. A series of simulations are conducted using three commonly used Reynolds-Averaged Navier-Stokes (RANS) based turbulence models. Noise amplitudes of up to 22% are evident with the largest observed for all solutions computed on unstructured tetrahedral grids, whereas computations on hexahedral and polyhedral grid structures exhibit substantially less noise. The Spalart Allmaras turbulence model is shown to be far less susceptible to noise levels than two other commonly-used models for this particular application and a typical noise sample is shown in Figure 2.

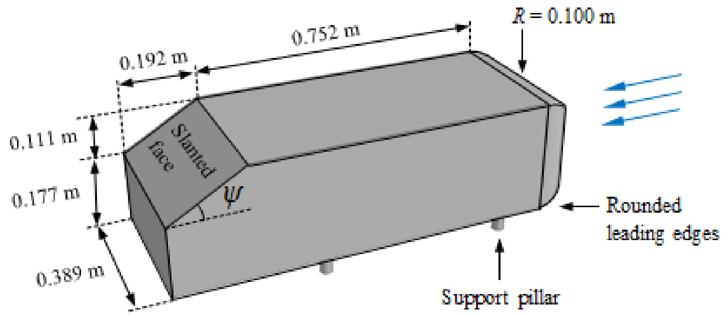

Figure 1: Illustration of the Ahmed body with relevant dimensions for a rear slant angle of $\psi = 30°$.

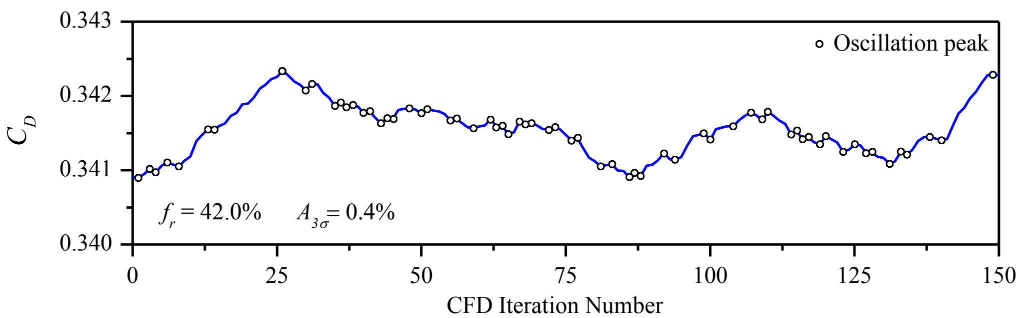

Figure 2: Sample of numerical noise exhibited by a Spalart Allmaras solution on a fine hexahedral grid.

The second part of the investigation considers multi-objective aerodynamic shape optimization for a coupled off-road vehicle and livestock trailer. Optimization is applied to a low-drag aerodynamic fairing which is parameterised in terms of three design variables. Moving Least Squares (MLS) metamodels are constructed from 50 high-fidelity CFD solutions for two objective functions. Subsequent optimization is successful for the first objective, however numerical noise levels in excess of 7% are found to be responsible for difficulties for the second one. Figure 3 shows the variation in both objective functions as one particular CFD solution develops, with a typical mixture of high and low-frequency modes present.

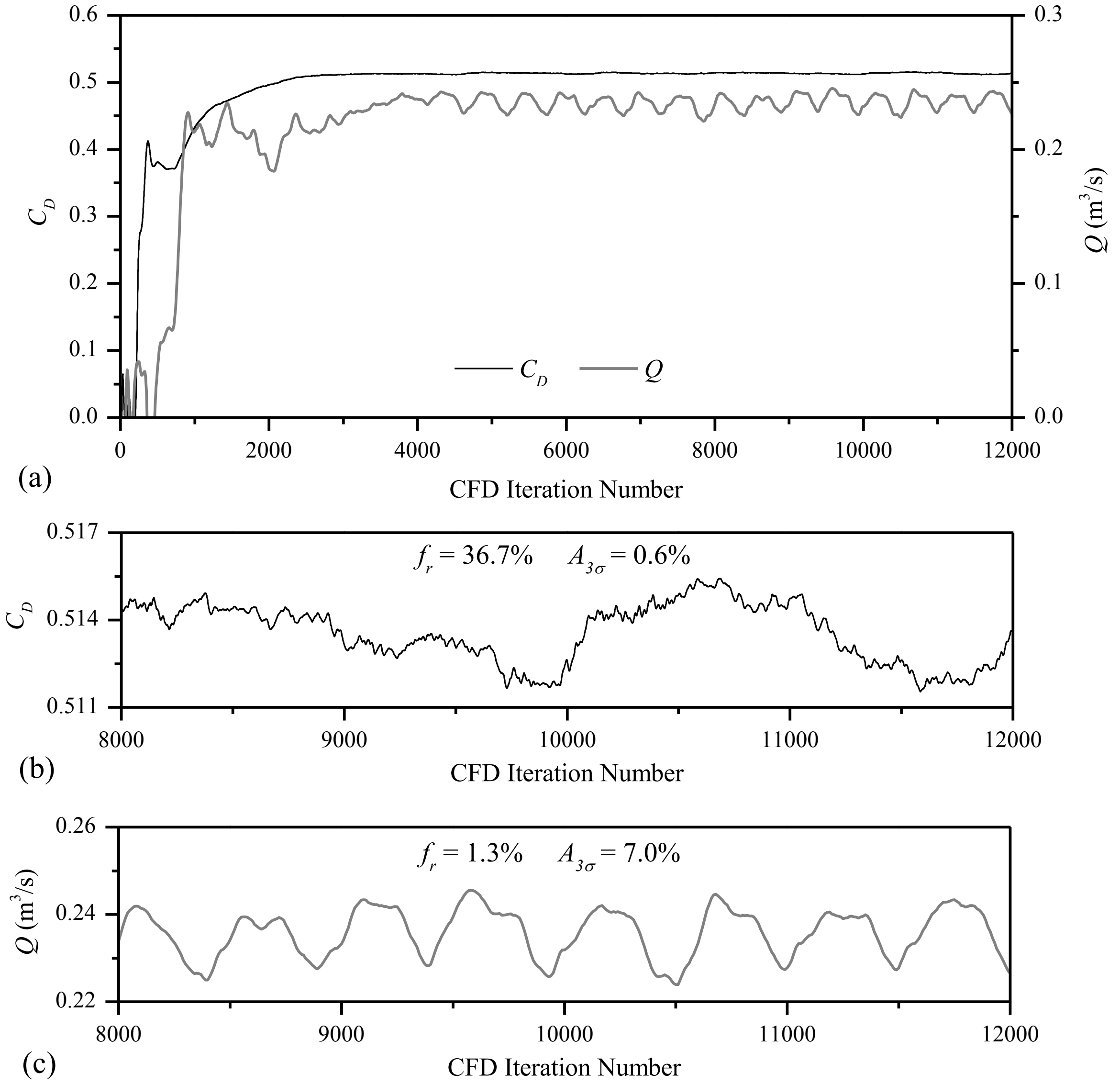

Figure 3: Plots showing typical noise levels for one CFD solution.

A revision to the problem reduces the amount of noise present and leads to success with the construction of a small Pareto Front. Further analysis underlines the inherent capability of MLS metamodels in dealing with noisy CFD responses. Suggestions are also made to improve the chances of success for future investigations.

Reduced Modal-Basis Techniques in Transient Thermo-Mechanical Topology Optimization

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ABSTRACT

Topology optimization supports engineers from many industries to design structures for complex problems. A major challenge in the precision industry is to reduce the thermal error [1]. Thermal error refers to the mismatch between real and predicted displacements and/or displacement differences between points of interest and sensor locations due to temperature fluctuations. Typically, the thermal error has to be reduced within a certain time frame of interest. Hence, the evaluation of the complex behaviour of the thermal error requires a transient thermo-mechanical model that must be used in topology optimization.

Gradient-based optimization algorithms need sensitivity information with respect to the design variables. In topology optimization, the adjoint variable method is used to compute economically the sensitivities for the large number of design variables (e.g. [2]). However, the adjoint method leads to reversed transient analysis for time dependent problems, which is undesirable for large scale problems (e.g. 3D topology optimization).

In this study, the transient responses of the thermo-mechanical system are described by the eigenvectors and corresponding eigenvalues of the thermal system (i.e., the thermal modes and time constants). As the transient behaviour is known for the modal representation, the reverse analysis can be avoided. The disadvantage is the computation of the thermal modes and, for the bigger part, the eigenvector derivatives. However, a high-quality approximation of the analysis may be sufficient.

In our approach, the response is expressed in terms of a relevance-based modal basis [3], which is determined considering three criteria: (i) modal excitation by the thermal loads, (ii) modal observability on the objective (i.e., thermal error), and (iii) modal participation within the time frame of interest. Then, we assume that the main features of the topological sensitivities are captured by the same modal basis.

Figure 1 presents the design case used in this study, which is inspired on an industrial application. Comparing the material layouts obtained by standard topology optimization (Figure 2) and those obtained using the proposed approach (Figure 3), it is seen similar topologies can be found. However, the thermal error (i.e. the objective) can be lower for material layouts obtained with a lower number of relevant modes compared with the thermal error of the reference design. Hence, the selection of the number of relevant modes to take into account is of great importance.

The modal representation is a good option to handle sensitivities with respect to topology variables for large scale transient problems.

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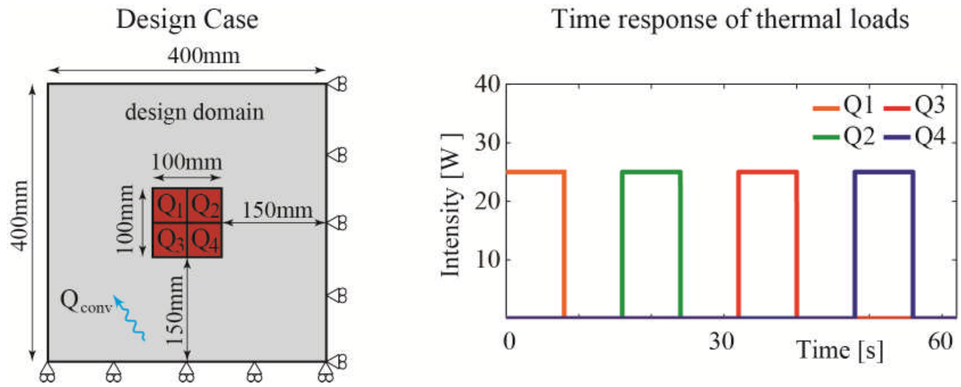

Figure 1 – The design case consists of an aluminium plate. Thermal loads are applied sequentially on the four quadrants Q1, Q2, Q3 and Q4, as indicated by the red squares. On each quadrant a 25W heat load is applied for 8 seconds, followed by an 8 second pause for each loading, as shown at the right. This could represent, e.g., a kind of measuring process. The plate is cooled from below, which is modelled by linear convective cooling with a heat transfer coefficient of 100 W/m2/K. The thermal error is measured as the absolute displacement of the centre point of a quadrant while loaded. The objective of the topology optimization is to minimize this thermal error.

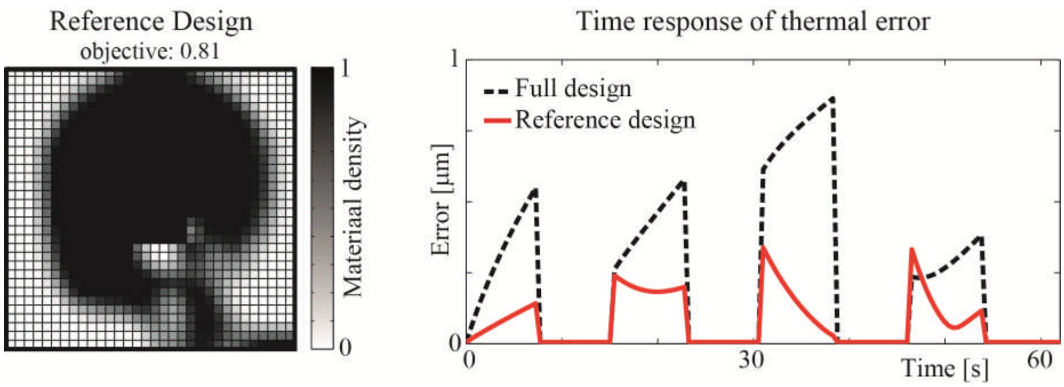

Figure 2 – Left, the resulted material layout for the topology optimization using a numerical integration in time. This topology can be seen as a reference. Right shows the thermal error as function of time.

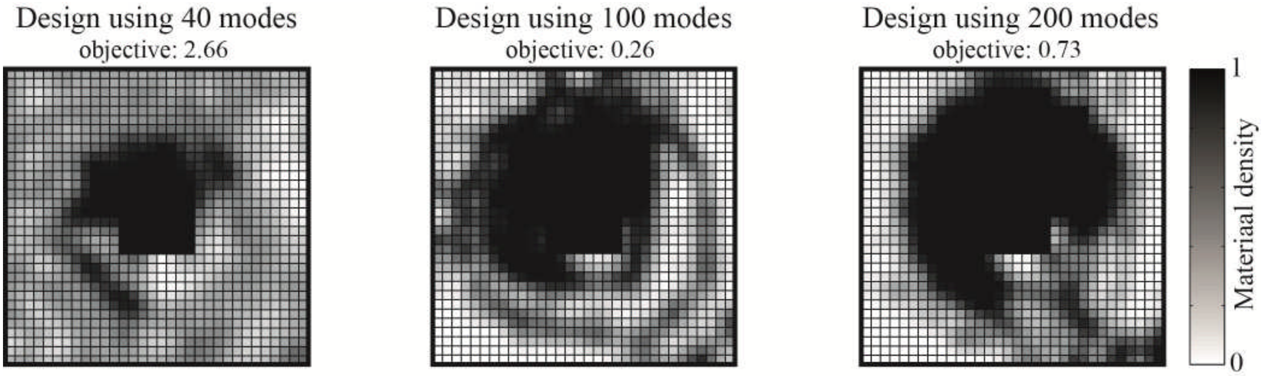

Figure 3 – Three designs obtained using different number of relevant modes. Increasing the number of modes results in topologies similar to the reference layout. But, the objective can be decrease more for other designs obtained with a lower number of modes.

Optimal Topologies of Extrusion Cross-sections for Crashworthiness

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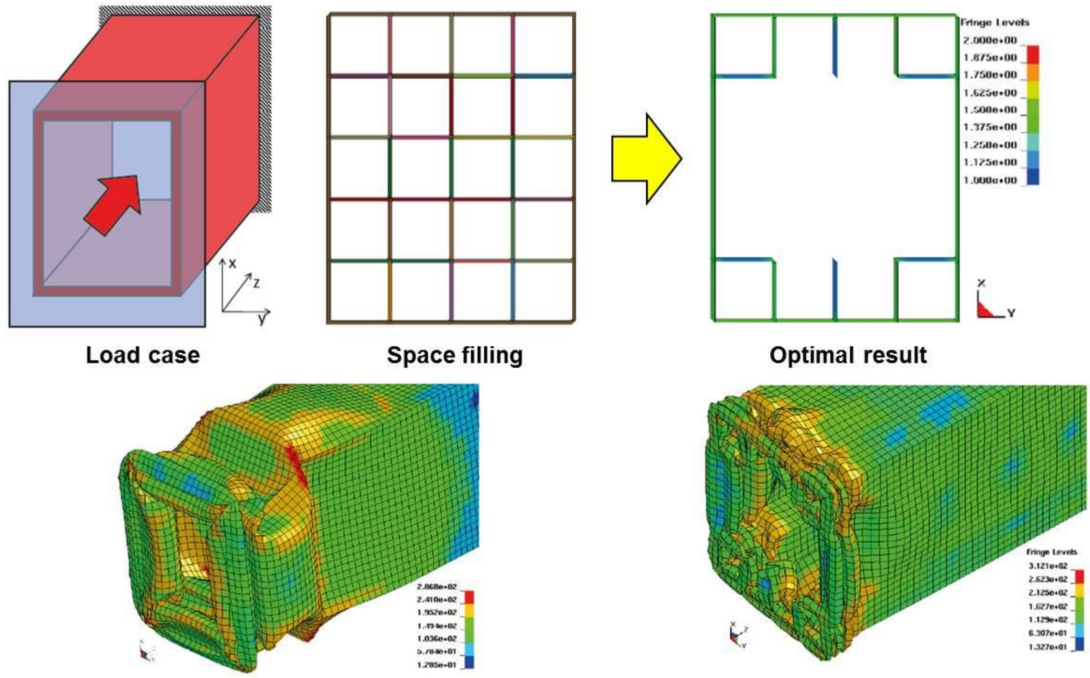
Keywords:
Topology optimization, crashworthiness, energy absorption, thin-walled structures, extrusion beams, ground structure approach, hybrid cellular automata.

While topology optimization is well established for structural problems with linear characteristics, it is still an open research field for highly non-linear cases like crashworthiness. To derive appropriate methods for the latter, it is necessary to distinguish on the one-hand side between different structural performance requirements: either the structure is designed for (i) high energy absorption or for (ii) high deformation resistance. For the latter case, several publications have proposed the usage of equivalent static loads, e.g. Volz (2011). Then classical topology optimization (e.g. Solid Isotropic Material with Penalization, SIMP) using linear elastic finite element methods can be employed. To minimize the compliance, the internal deformation energy is distributed as homogeneously as possible. While this works for the high deformation resistance case, it is not appropriate for structural optimizations with high energy absorption as objective. Here strong plastic deformations with relatively high forces and eventually failure of the material have to be considered. Normally, metal structures with thin-walled and hollow cross-sections are optimal. Extruded beams are advantageous with interior reinforcements (multi-cell cross-sections). Their reaction is based on the formation of a plastic collapse mechanism with patterns of explicit plastic hinge lines. In these cases, the internal deformation energy is concentrated in these hinge lines such that a homogeneous distribution of such energy should not be taken as optimization objective. Most of the existing methods for topology optimization have not solved this issue. In addition, most of the approaches are based on two- or three-dimensional voxel techniques, which will never result in thin-walled beam structures. This argumentation might be slightly different when non-metal structures are discussed. For example composite structures show often superior energy absorption by progressive local failure. Catastrophic failure due to sudden delamination of the layers should be avoided. This request for robustness is also difficult to include into the topology optimization studies.

On the other-hand, topology optimization problems for crashworthiness have to be categorized into cases where (iii) the principal structural concept, i.e. the load paths related to the impact cases, has to be derived and cases where (iv) a component has to be optimized in one of these load paths. For the load path topology, the optimal assembly of structural components in the design space defined by the packaging is searched for. For component topology, reinforcement patterns, crush initiator patterns or tailored blank patterns have to be determined by the optimization process. For the former, ground structure approaches are an option where structural elements are successively eliminated or introduced. In particular the modular techniques based on a library of parameterized generic components, which can adapt to the current structural configuration, are attractive. This is for example offered by the software SFE CONCEPT, e.g. Duddeck (2012).

Considering the situation described above, this paper presents a new approach, which addresses topology optimization for crashworthiness focusing on (i) energy absorbing areas and (iv) component topology. It is based on non-linear crash simulations of thin-walled structures. Hence the correct energy absorption mechanism (plastic hinge formation) is taken into account. The hybrid cellular automata (HCA) approach proposed by Patel (2007) for voxel techniques, using homogeneous energy distribution as objective, is transferred here to a macro-element (or ground structure) approach replacing voxel cells by larger thin-walled structural elements allowing localized plastifications. To the authors' knowledge, this enables for the first time the derivation of optimal reinforcement patterns of thin-walled extrusion beams. Examples of axial and oblique impacts illustrate the potential of this approach, see the example of an axial impact given in Fig. 1. Details of the method have been published in Hunkeler (2013).



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Additive Manufacturing and Topology Optimization

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Symbiosis and Challenges

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Abstract

After pioneering research, topology optimization gained major interest from the structural optimization community. In topology optimization, the layout of a structure is created by an algorithm that optimizes a response, while satisfying certain constraints, e.g. eigen frequencies, mass etc. In fact, topology optimization seeks an optimal placement of material, leading, besides the layout of a structure, to shape and dimensions as well. Early industrial adopters can be found in automotive and aerospace industry.

Topology optimization is typically based on application of finite element models. Consequently, the topological description is mostly directly connected to the individual finite elements by the introduction of a virtual density for each element. This leads to a voxel-based design representation. Computational efficiency requires so-called adjoint design sensitivities to be used in combination with a gradient-based optimizer.

Topology optimization may lead to very complex 3D structural designs in terms of shape and topology. Particularly in 3D settings these design are superior to manually created design. The advantages will be even more prominent in 3D multidisciplinary settings. The limitations of classical production techniques require in

Figure 1: Complex alternated gripper. This complex 3D compliant mechanism was designed with topology optimization and produced using additive manufacturing.

general a major engineering effort to translate a topology-optimization based design into a real product. This situation changes dramatically if modern additive manufacturing techniques can be used. These techniques can easily produce very complex 3D products. Moreover, the voxel-based design representations as used in

topology optimization are extremely suited to be directly combined with the additive manufacturing machines. See Figure 1 for an example.

In this presentation we will briefly introduce topology optimization and its potential in combination with additive manufacturing. In this discussion we particular focus on mechatronic precision applications. In this context, we shall review the state-of-the-art in topology optimization and identify the associated main challenges. Aspects that will be highlighted are transient response functions which reflect the performance of precision systems, manufacturing constraints, process-product modeling, multiple physical domains, resolution and computer time, artifacts, nonlinearity, aspects of control and CAD interfacing.

Weight optimisation of the truck cabin part from thermoplastic composites

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Abstract

Regarding high limit stress which is close to steel yield strength, thermoplastic composites with woven glass/fibre (GF) and polypropylene (PP) fibres have demonstrated exploitable potential as valuable alternative to sheet metal providing good strength/weight performance ratio. At the same time the thermoplastic composite mechanical properties are similar as traditional glass fabric laminates with epoxy resin matrix however manufacturing time advancing to extremely fast by hot mould pressing.
In current research topology optimisation in conjunction with parametrical optimisation are being investigated in order to find the most efficient material distribution over the car part surface which is made of thermoplastic composite. To create topology plots of material volumes, numerical model in ANSYS finite element code have been created taking into account necessary loads and boundary conditions. In this case dominating load is forward drag force because truck driver seat is attached on top of the composite plate and fixed in four corners. As the result volume distribution plots like shown in Figure 1 are acquired where it is clearly seen that largest material volume should be located in the corners under fixation places.

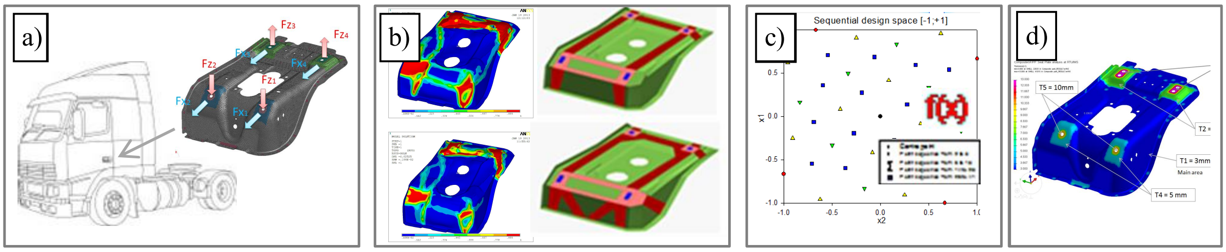

Figure 1. Workflow of the optimisaton process. a) load case definition based on application case in truck cabin; b) topology optimization of the part and interpretation of the results; c) parametrical optimization of the discrete number of the variable thicknesses; d) final structure

However acquired topology plot can`t be directly used for choosing correct local reinforcement thicknesses because manufacturing constraints often limit the physical realisation of the optimal model and trade-off between optimality and manufacturability should be found. Interpretation of topology optimisation results often bring disagreement between scientists and engineers, therefore current industrial standard for processing topology optimisation plots is still a manual interaction by engineer.
For seat plate composite part topology optimisation results have been used as early input data for location of the most critical areas. As the result regular square patches have been assigned for areas around support points due of difficulties manufacturing variable thickness woven fabrics. For final design of the local reinforcement thicknesses parametrical optimisation has been performed where thicknesses for the patches are necessary variables.
As the result of applying combined optimisation method self-weight of the structure have been significantly reduced. The weight of the optimised composite part is only 35 % of the reference steel part.

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Optimal design of composite structures with curved fiber trajectories

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I. Introduction

IT is well known that using curvilinear fiber paths can significantly improve the structural performances of composite structures [1,2]. The fiber placement technology allows today to manufacture composite structures with such curvilinear fiber paths, should the component be flat or present a curvature. Several algorithms are available to simulate the fiber trajectories, see for example [3-5]. They are most of the time based on a reference fiber direction, which is translated, based on complex geometric equations, in order to provide tows as parallel as possible to each other. Most of these algorithms fail to provide such parallel courses, and in practice overlaps and gaps appear between adjacent tows, leading to over-thickness or small voids, where delamination is prone to occur and where the material allowables are difficult to estimate. Some other algorithms are demonstrated only for simple almost flat structures [3] or for specific geometries [4].

In this paper, the fiber trajectories are computed over a 3D surface using the fast marching based method [6] presented in [7]. This method assumes that the parallel courses of the fiber placement machine are the positions of a propagating wave front over the surface. The wave front is assumed to be infinitely long in order to define courses which cover the whole surface. The general 3D surface is defined by a 3D mesh. A reference fiber is defined over the mesh. It represents the general shape of the fiber over the surface and it is the initial position of the wave front. The Eikonal equation is solved over the 3D mesh, with the reference fiber as an initial condition, to compute the travel time of the wave front at the nodes of the mesh. A modified fast marching method is proposed in the paper [7] for the case of an infinite wave front. The position of the wave front, which is the fiber course, is obtained from iso-values of the computed travel times (Figure 1).

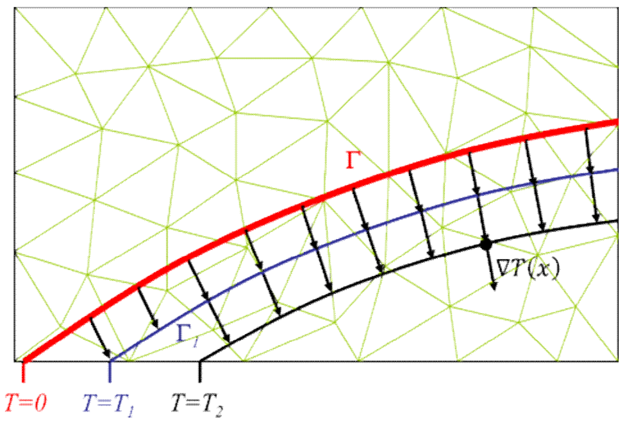

Figure 1. Reference fiber and solution of the Eikonal equation on the mesh

This new approach is then used to solve optimization problems, in which the stiffness of the structure is maximized. The design variables are the parameters defining the position and the shape of the reference curve.

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The shape of the design domain is discussed, regarding local and global optimal solutions. Different optimization methods are compared on several applications. A discussion on the sensitivity analysis for gradient-based methods is proposed. The benefit in using such a parameterization is discussed based on a comparison to a solution relying on local optimal orientations in the structure. A first solution [8] is presented in Figure 2.

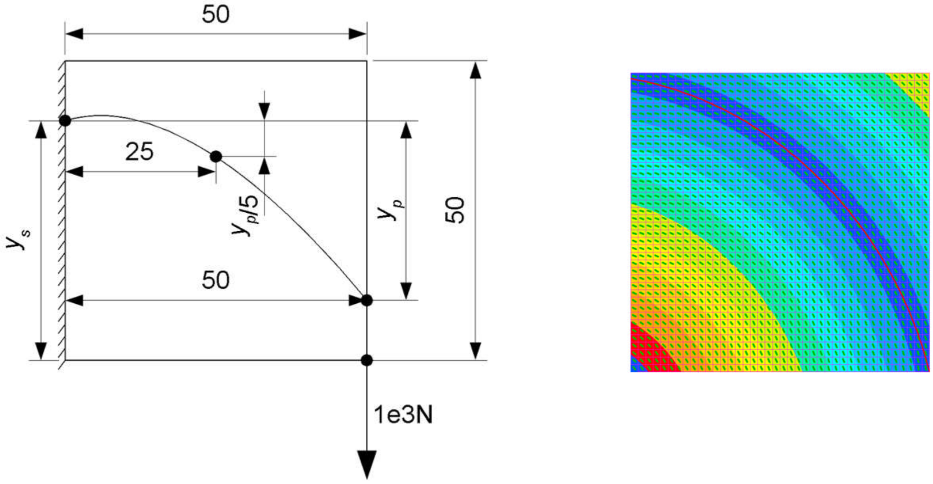

Figure 2. Parameterization and solution

Acknowledgements

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Mid-Range Approximations in Sub-spaces for MDO problems including disparate attribute models

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The current work is an attempt to create an efficient MDO framework for solving optimisation problems including a series of high speed, non-linear explicit models (i.e. crashworthiness assessments) that depend only on restricted subsets of the total design variable set as well as much less complicated (i.e. linear static) models that depend on (almost) all design variables.

Responses from crashworthiness analysis can be highly non-linear and even discontinuous as some of the contacts may be active in only certain regions of the design space. A simple test case is shown in Figure 1. The left hand side load case is a cylinder impacting a metal sheet beam. The response is the resulting maximum effective plastic strain. On the right hand side is a torsion load case where the beam itself is twisted with a certain moment and the twist is measured. The design variables are the two middle panels on the top side of the beam.

The response surfaces in Figure 2 highlight the difference in complexity for the two load cases. The left hand side plot shows a highly non-linear equivalent plastic strain response resulting from the explicit non-linear impact simulation while the right hand side plot shows a smooth response surface resulting from the linear static torsion load case.

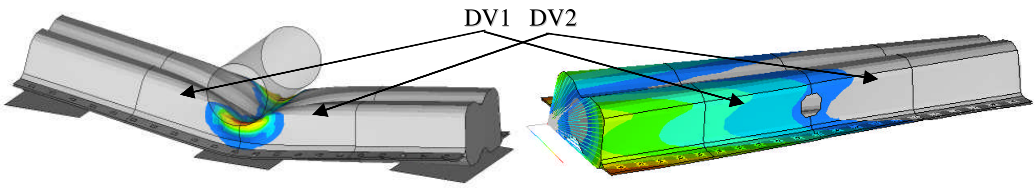

Figure 1. Left: Impacting cylinder (explicit) load case. Right: Static torsion (implicit) load case.

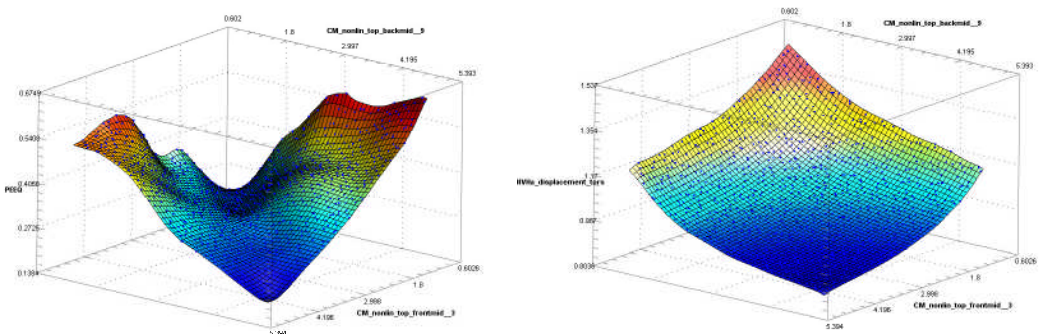

Figure 2. Left: Equivalent plastic strain (impact load case). Right: Twist (static torsion load case).

Because of the considerable disparity in complexity of the models and resulting response surfaces, one can draw the conclusion that there should be different procedures for handling the different types of models within the MDO framework. The more complex impact load case presented would need more DOE points to approximate the response behaviour than the more simple static torsion load case.

While the crash response is highly non-linear and requires a higher density test plan, in many cases they might be affected only by a subset of the total set of design variables. Consider the simple beam model in Figure 1. The static torsion subcase will be affected by (almost) the entire structure but the crash response is mostly affected by the two design variables that were chosen. By using engineering knowledge and variable screening methods, as described by Tu and Jones (2003) the dimensionality of each test plan can be brought to its minimum.

The multipoint approximation method, as reported by Toropov (1989), Toropov (1992) and Toropov et al. (1993) is an iterative optimisation technique based on mid-range approximations built in trust regions. A trust region is a sub domain of the design space in which a set of design points, treated as a small-scale design of experiments (DOE), are evaluated. These and a subset of previously evaluated design points are used to build approximations of the objective and constraint functions that are considered to be valid in the current trust region. The trust region will then translate and change size as the optimisation progress. The trust region strategy has gone through several developments to account for the presence of numerical noise in the response function values, see van Keulen et al. (1996), Toropov et al. (1996) and occasional simulation failures (also termed domain-dependent calculability of the response functions), Toropov et al. (1999). The mid-range approximations used in the trust regions, as originally suggested by Toropov (1989), are intrinsically linear functions (i.e. nonlinear functions that can be led to a linear form by a simple transformation) for individual sub-structures, and assembly of them for the whole structure. This was enhanced by the use of gradient-assisted approximations (Toropov et al., 1993), use of simplified numerical models that is also termed a multi-fidelity approach (Toropov and Markine, 1996) and the use of analytical models derived by Genetic Programming (Toropov and Alvarez, 1998). The most recent development (Polynkin and Toropov, 2012) involved the use of approximation assemblies, i.e. a two stage approximation building process that is conceptually similar to the original one used by Toropov (1989) but free from the limitation that lower level approximations are linked to individual substructures.

The moving least squares method was proposed by Lancaster *et al.* (1981) for smoothing and interpolation of scattered data and later used in the mesh-free form of the FEM (Liszka, 1984). As described by Choi et al. (2001) it can be used as a technique for surrogate modelling and used in MDO frameworks. The moving least squares method is a weighted least squares method where the weights depend on the Euclidian distance from a sample point to where the surrogate model is to be evaluated. The weight value for a certain sample point decays as the distance increases. Describing the weight decay with a Gaussian function tends to be the most useful option even though many others have been evaluated by Toropov *et al.* (2005). As demonstrated by Polynkin *et al.* (2010) the cross validated moving least squares method can be used both for design variable screening and for surrogate modelling.

In the presented research the multipoint approximation method is extended to use local test plans and moving least squares approximations built in different subspaces of the total design variable space in an attempt to create an efficient MDO framework for incorporating crashworthiness assessment in MDO.

The presentation will describe the optimization process in which to deal with the curse of dimensionality different parameterization approaches are used for the computationally heavy responses (e.g. crashworthiness-related) and the lighter responses (e.g. buckling, global stiffness, etc.). To keep such an MDO problem solvable the heavy responses are represented by a smaller number of design variables, and the lighter responses may require many more design variables. This way the approximation building DOEs are separated, i.e. have a small number of DOE points for the crashworthiness responses and many more for the lighter responses. Finally, all the sub-spaces corresponding to all the disciplines are combined into the total design variable space of the whole problem and an optimization problem is iteratively solved in each of the trust regions of the combined space.

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Multiobjective and Robust Design Optimization of an Active Solar Energy System

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OPTIMUS is a Process Integration and Design Optimization (PIDO) platform that allows to integrate and combine any scientific or engineering software tool into a single simulation workflow. Once a workflow is defined, Optimus orchestrates the simulation process to automatically explore the design space and identify the optimized and robust solution.
In this paper we first present how Optimus can easily integrate with any external software, and in particular we show the integration with Scilab and all its capabilities as a computing environment for engineering and scientific applications. The easiest way to integrate a software environment like Scilab is by means of a so-called "User Customizable Action" (UCA). Any such UCA can then be included in a (multidisciplinary) simulation workflow using Optimus' graphical drag and drop interface.
To demonstrate the benefit of having an independent platform integrating various software, we present an example that optimizes the performance and costs of an active solar system. In this example, a Scilab script is used for "Design and modeling the φ-f chart method for active solar energy systems" [1] while an Excel spreadsheet is calculating the total cost of the system according to the selected components. This represents a multiobjective and multidisciplinary problem in which - by changing parameters such as the collecting areas of panels, type of supports … - Optimus automatically identifies all Pareto-optimal solutions.
Fig.1 depicts the Optimus workflow that runs the Scilab and Excel scripts and macro in batch mode with their own UCA. The workflow is changing each time the values of the input parameters and it is finding the best configuration that minimizes the global cost while maximizing the efficiency of the system. To deal with multiple objectives, discrete variables and non-linear responses a multiobjective particle swarm optimization (mPSO) algorithms is used. The mPSO algorithm available in Optimus efficiently handles high-dimensional optimization challenges, supports parallel execution of experiments, and delivers a highly accurate optimal Pareto front.

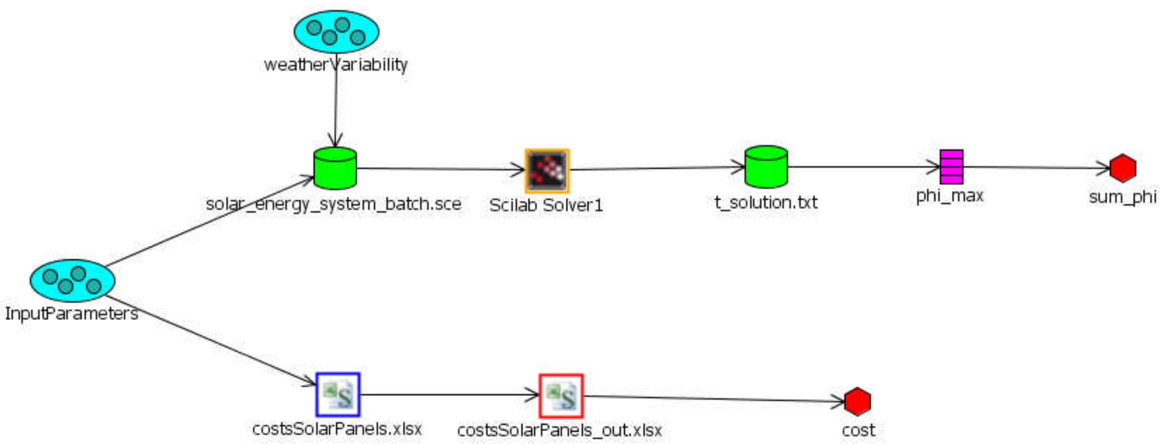

Figure 1 Example of an Optimus workflow: identification of optimal parameters to maximize performance while reducing costs.

As a second part of the work, a robust design optimization of the solar energy system is performed. The uncertainties related to the design of a solar thermal energy system may derive from tolerances in all components and in any external events that are not completely known.

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Deterministic approaches to optimization do not consider the impacts of such variations and, as a result, design solutions may be very sensitive to these variations and result on an average performance that differs from the expectations. In practice, not taking into consideration such variations usually implies a considerable over-sizing of the system resulting in an overall increase in costs.

In this paper we apply the uncertainty to the irradiance data taking into account that the amount of solar radiation that reaches the ground not only depends on the geographical location and the yearly apparent motion of the sun, but also on the climatic conditions and the cloud cover of the sky.

The cloudiness is indeed the main factor affecting the efficiency of solar energy panels and the clearness index should then be treated as a random variable with its appropriate probability distribution function (PDF) [2-5]. The distribution that is taken into account in Optimus for robustness can be either theoretical or based on historic recordings. Once the PDF is defined, the probability of failure and/or the reliability index (expressed in number of standard deviations) can be easily computed for any optimal configuration.

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Study of stiffener design for the Pantheon's dome using the Isolines Topology Design method

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Shell structures are extensively used in architecture and in aeronautical, civil, marine and mechanical engineering. The last few decades have seen remarkable advances made in the optimization of their size and shape. Although during the same time topology optimization has seen great advances, there seem to have been few attempts at applying topology optimization to the design of shells. This may be attributed to topology optimization generating cavities inside and modifying the boundary of the shell structure. Isolines Topology Design (ITD) is an iterative algorithm which uses the contour map or isolines of the design criterion for a structure in order to determine its optimal topology. This paper presents an enhancement to the ITD method which allows the design of the stiffeners in shell structures. In order to obtain the stiffener layout, the shell is modelled using overlapping layers of shell finite elements with shared or coupled nodes in which the shell structure (base material) layer is not optimized and the others generate the stiffener locations. Nowadays, the Pantheon's dome (located in the centre of the city of Rome) is still the world's largest unreinforced solid concrete dome and its 43.4 metres in diameter still impresses the structural engineers. In this work, the shell dome of the Pantheon was chosen to demonstrate the applicability and effectiveness of ITD algorithm to obtain the exact stiffener location, without the need to interpret the resulting layout, using three layered models.

I. Introduction

A shell is a three-dimensional (3D) structure bounded primarily by two arbitrary curved surfaces a relatively small distance apart, Zingoni [1]. Such structures are used extensively in architecture and in aeronautical, civil, marine and mechanical engineering [2-4]. They have been optimized in one of three ways: 1) The thickness of the shell was optimized, whilst maintaining the original shape of the structure; 2) The shape of the shell was optimized by moving the control points which defined it, but keeping its thickness unchanged; and 3) The topology of the shell was optimized using topology optimization where both its shape and thickness could be modified. The application of topology optimization to shell structures has been the least researched of the three. This can be attributed to two consequences of topology optimization: 1) Cavities are introduced into the structural domain; and 2) The perimeter of the structure can be significantly modified. Since a primary use of shell structures is to cover, shield or enclose a space or volume, the two consequences mentioned would severely affect the applicability of a topologically optimized shell structure. A reason for using shell structures is that they are lightweight and can be easily manipulated into the desired shape. But they suffer from poor overall stiffness, something which can be addressed by the strategic addition of stiffeners.

The aim of this paper is to apply the Isoline Topology Design (ITD) method [5] to the problem of topology design of stiffeners for shell structures. The novelty of this work is that the shell structure is modelled using overlapping layers (overlapping-shell model) of thin-shell FE, where the nodes of the overlapping FE are shared or can be coupled using multi-point constraints. One of the layers represents the shell structure or base material and the other provides the stiffener location. Note that the base material is not subjected to the optimization

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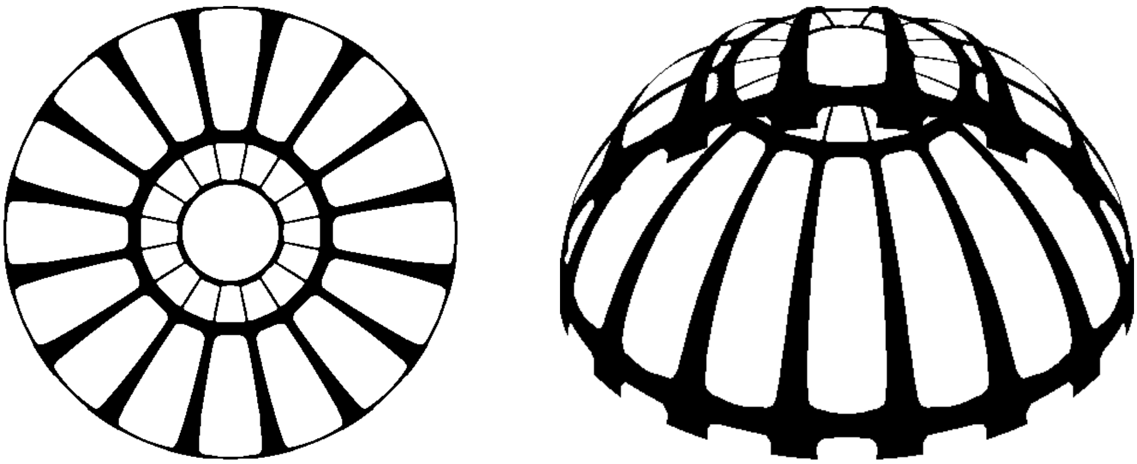
process. In order to demonstrate this approach, the modified ITD method is briefly explained and applied to study of the Pantheon's dome using three layered models.

II. Example

The design domain consists of a hemispherical shell 43.4 meters wide and thickness t = 2 m with an opening of 9.1 meters in diameter at the top (Fig. 1). The design domain is subjected to a self-weight load equal to initial weight. The bottom circumferential boundary of the hemisphere is just supported (i.e. only vertical displacement is not allowed). The material properties (common steel) used to obtain stiffener layout were: elasticity modulus 210 GPa, Poisson's ratio 0.3 and mass density 7850 Kp/m^3. The FE used was the ANSYS SHELL63 [6], which is based on the Kirchhoff-Love theory. Only a quarter of the domain was analysed using a regular rectangle mesh. Note that, full design domains are shown here to obtain clearer results.



(a) **(b)**

Figure 1: Stiffener layout design using single-layer model: (a) Top; (b) Isometric views

The resulting topology using the single-layer model (Fig. 1) reveals that:

1. The design generated using ITD produces a stiffener layout in good agreement with the original solution (Pantheon's dome). However, the number of vertical and horizontal ribs (16 and 3, respectively) is significantly lower.
2. The angle which divides between tensioned and compressed parallels is less than 52º.
3. The horizontal rib located around the base (tension ring) is considerably larger than the others since the horizontal component (outward thrust) near of dome base is highest.

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Sense and nonsense in structural topology optimization

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Introduction

The aim of this paper is to review critically the history of structural topology optimization, with particular attention to controversial issues in the literature. It is also attempted to trace possible sources of errors, and to put up arguments in support of the Authors' opinion on various issues. The time span of the survey is from the beginning of the 20th century to present day.

Exact truss topology optimization

Structural topology optimization started with the pioneering paper of the brilliant Australian inventor Michell (1904) from Melbourne, who laid down the foundations of exact analytical truss topology optimization. Basically, Michell's theory says that for stress-based truss volume minimization, the strains in the bars must be imbedded in a virtual strain field over the structural domain, such that (i) along optimal bars (of non-zero cross sectional area) the strains take on a constant value, and (ii) along any other line segment the strains are smaller than or equal to that reference value. It was implied by Michell that the above optimality criteria are also valid for different permissible stresses in tension and compression.

Rozvany (1996) (i) pinpointed a flaw in Michell's proof, (ii) stated the range of validity of Michell's optimality conditions, (iii) derived the correct optimality criteria for problems outside this range, by using three different methods, (iv) presented a simple example showing that the modified optimality criteria result in a much lower truss volume than Michell's original ones, and (v) pointed out which examples in Michell's paper represent non-optimal topologies. Further details, including diagrams with illustrative examples, will be shown in the lecture.

In his outstanding book, Hemp (1973) introduced his orthogonality principle: 'If a pair of tension and compression members meet at a point, they must be orthogonal … no other member can be coplanar with them'. It has been explained in several publications (e. g. Rozvany 1997) that Hemp's orthogonality principle is valid only if a compression and a tension member meet at a point that is on the interior of a so-called T-region, but if the intersection is on the boundary of two R-regions, then this principle becomes invalid. The concept of optimal regions (such as R, T, S and O-regions) will be explained in the lecture.

Exact topology optimization of grillages (beam systems) and reinforcement layout

The theory of optimal grillage topology was developed by Rozvany's research group in Melbourne, some reviews can be seen in papers by Rozvany and Hill (1976) and Prager and Rozvany (1977). It is not so well known, however, that the basic idea of the so-called adjoint strain fields for grillages came from Morley (1966), who set up a theory for the mathematically similar problem of optimal reinforcement topology in concrete slabs. It is interesting to note, that Morley declared that a solution satisfying his optimality criteria for clamped corners does not exist, but Melchers (Lowe and Melchers 1972) has found such a solution.

There was a short controversy about Melchers using slope-discontinuous adjoint displacement fields (see a review in Rozvany's 1976 book, pp. 265-268), but this was quickly resolved after Melchers discovered many optimal topologies by first assuming slope-discontinuities, and then eliminating the discontinuity by optimization. After important contributions by Melchers and Hill, grillage optimization became the first class of plane research problems, for which the exact analytical solution is known for almost any boundary and loading condition.

'Moment balancing method'. Considering concrete slabs, it was claimed in several publications by Brotchie (e.g. 1962) that the optimal solution for orthogonal reinforcement or tendons is given by the moment components M_x and M_y that 'satisfy the elastic plate equation' (i. e. the usual biharmonic equation). After several years of controversy, Brotchie (1967) put forward a 'formal proof', saying that by certain energy theorems, the elastic solution corresponds to energy minimization, and 'minimal potential energy thus results in (i) maximum stiffness and minimum deflection; and (ii) minimum material quantities'. Needless to say, Brotchie mixed up energy theorems for the analysis of isotropic plates with the optimization of reinforcement in slabs, or the mathematical equivalent problem of grillage topology optimization.

'Monte-Carlo approach' to topology design. This was one of the extreme misconceptions in reinforcement optimization, suggested by Muspratt (e.g. 1970, see Rozvany 1971), who claimed that reinforced concrete slabs should be optimized by the 'Monte-Carlo approach', in which the length and location of each bar is assigned in a random fashion. He even showed such a design in the above publication. Muspratt tried to justify his method by stating that location and length of bars are uncertain quantities, having a certain probability distribution. Although this is correct, and Monte-Carlo simulation is a very useful method for random sampling and for generating probability distributions numerically, its use for randomly deciding on the value of design parameters is entirely unjustified. This will be demonstrated in the lecture.

Numerical (discretized) methods in structural topology optimization

Gradient-type or sensitivity-based methods. The presently most popular numerical, FE-based topology optimization technique is the **SIMP method**, which was developed in the late 1980s. It is sometimes called “material interpolation”, “artificial material”, “power law”, or “density” method, but “SIMP” is now used fairly universally. The term “SIMP” stands for Solid Isotropic Microstructure (or Material) with Penalization of intermediate densities. The basic idea of this approach was proposed by Bendsøe (1989), whilst the algorithm was developed later (e. g. Zhou and Rozvany 1991), and the term “SIMP” was coined afterwards (Rozvany, Zhou and Birker, 1992).

Zhou implemented the SIMP method in the OptiStruct software of Altair, which is used extensively for topology optimization by the car and aeroplane industry. Most other commercial software also employs the SIMP method. However, the full acceptance of this method in academic and research circles is the merit of Sigmund, who is also a leading researcher in the area of various improvements and applications of this algorithm.

So-called ‘hard-kill’ or ‘sudden-death’ methods introduce finite changes in a design on the basis of certain heuristic conditions, which are somewhat similar to gradient criteria . One of these is inappropriately called **ESO** (Evolutionary Structural Optimization), because “evolutionary” usually refers to Darwinian processes (as in genetic algorithms), and “optimization” implies computation of a truly optimal solution, which is not necessarily the case with ESO. As appropriate term for this method “SERA” (Sequential Element Rejections and Admissions), has been suggested by Rozvany and Querin (e. g. 2002).

SIMP vs. ESO. In a Brief Note, Zhou and Rozvany (2001) used a simple example to demonstrate the failure of the ESO method, and also showed an intuitively good solution for a volume fraction of 40%. In a very interesting paper, Stolpe and Bendsoe (2007) confirmed global optimality of that solution by both a non-linear branch and cut method and by simple enumeration.

In a long forum article, Rozvany (2009) compared the SIMP and ESO methods in considerable detail, and also suggested improvements of the ESO method under the name SERA. Some of these have already been considered in earlier papers by Rozvany and Querin (e. g. 2002). In response to the above observations by Zhou and Rozvany (2001)and Rozvany (2009), Huang and Xie (2008, 2010) published some highly constructive comments in two papers. As a result of a highly productive exchange of ideas, ESO could become a useful alternative to the SIMP method.

Another promising gradient type technique is the **level-set method**, which is subject to intensive research at present, but has not quite reached yet the stage of industrial application. Its advantage is the generation of smooth boundaries, but at present it is computationally less economical than SIMP, and the solution can depend on the choice of the level set function. It can be effectively combined with topological derivatives.

Gradient vs. non-gradient methods

Sigmund (2011) pointed out that gradient type topology algorithms can solve problems with up to millions of variables using a few hundred (and some commercial codes even less than 50) function evaluations. On the other hand, non-gradient methods need typically over 20,000 function evaluations even for very coarsely discretized problems.

Sigmund (2011) convincingly dismissed arguments for non-gradient methods, that (i) they lead to better optima being global search methods, (ii) they provide discrete designs which are better than grey-scale designs, (iii) they are easy to implement because they do not need any gradients, and (iv) their advantage is that they run easily on parallel computers. Sigmund also summarized the disadvantages of non-gradient methods, including the fact that they cannot be used for method verification by gradually refining the finite element mesh for comparison with the analytical solution. However, Sigmund remarks that non-gradient methods may be appropriate for some very special problems with many local minima or disjoint design domains.

It is to be remarked that for certain classes of problems, some gradient methods can handle billions of variables (e. g. Sokół 2011). Le Riche and Haftka (2012) published a response to Sigmund's (2011) article, in which for non-gradient methods they use the term 'global optimization methods', although they concede that theoretical global convergence proofs for these methods have little practical significance. As editorial guidelines they propose that such papers should contain a pseudo-code or mathematical formulation devoid of any metaphor, and the difference from existing metaphorical optimization methods should be clearly explained.

Multi-load and probabilistic topology optimization

The most recent, but completely one-sided controversy has arisen from a paper on probabilistic topology optimization by Rozvany and Maute (2011). It was shown by the above authors, that the exact optimal topology for a probabilistic compliance problem is a symmetrical two bar truss, whose optimal geometry (the inclination of the bars) is given by a very neat closed form solution. Rozvany and Maute also proved that the above probabilistic problem can be converted into a deterministic one with two alternative load conditions, for which the analytical solution is the same two-bar truss. The above exact solution has been confirmed numerically by sixteen authors.

In the meantime, Logo put in a Discussion on the paper by Rozvany and Maute (2011), and implied in his rather incoherent text certain shortcomings in the above paper. Amongst others, he stated that the deterministic equivalent of the Rozvany-Maute problem must consist of three load conditions, and that the optimal topology for that problem consists of three bars (instead of two). The above claims were disproved in an Authors' Reply (Rozvany and Maute 2013), and in greater detail in a longer paper by Rozvany, Pomezanski and Sokół (2014). It was shown that the equivalent deterministic problem consists of two loading conditions.

Moreover, they pointed out that Rozvany and Maute (2011) optimized the truss considering all possible geometries and topologies, whilst Logo only considered two and three-bar topologies having a given (non-optimal) geometry, with an fixed angle of 30% between the bars and the symmetry axis. the latter is an entirely different, and much simpler problem.

The optimality of the two-bar solution was shown by three different methods. First, exact optimality criteria were used to derive the optimal solution considering all possible topologies. Second, three-bar trusses were optimized analytically for the above problem, showing that the optimal solution turns out to be a two-bar truss. Finally, analytical solutions for the global, two-bar optima were compared with solutions having a fixed (non-optimal) geometry (as in Logo's solutions), and it was found that the latter never gives a lower volume than the former (see Fig. 1 below).

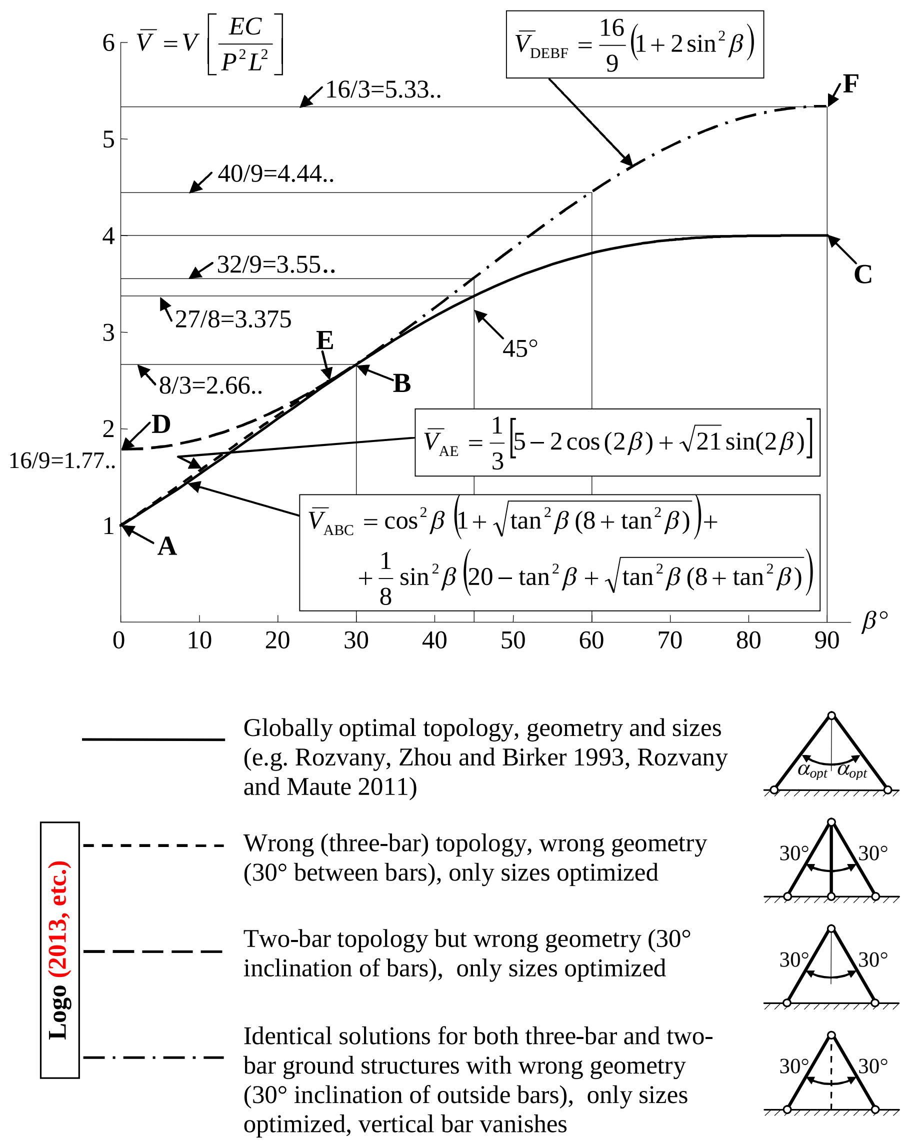
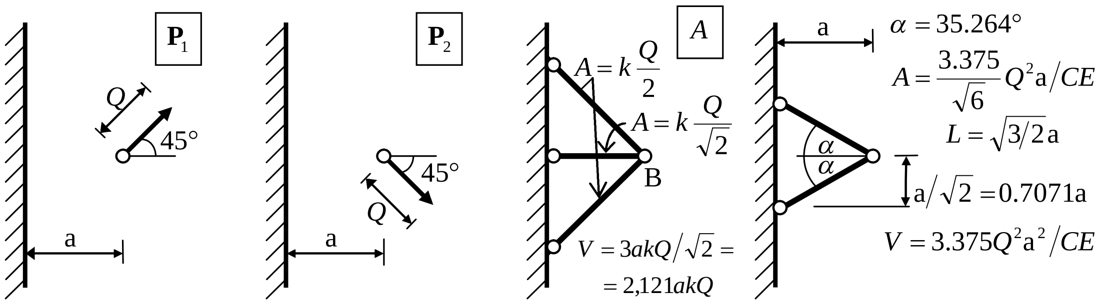


Fig. 1. Comparison of optimal and non-optimal solutions for the Rozvany-Maute (2011) problem.

In his Discussion, Logo referred to a paper by Nagtegaal and Prager (1973), saying that in it the optimal solution consists of three bars. It was pointed out in the Authors' Reply (Rozvany and Maute 2013) that The Nagtegaal-Prager paper considered plastic design, whilst the Rozvany-Maute (2011) study dealt with elastic design. The solution for these two problem classes are usually quite different, as is shown in Fig. 2 below.



(a) (b) (c) (d)

Fig. 2. (a) (b) Loading conditions, (c) optimal plastic design, (d) optimal elastic design

Nevertheless, the authors are grateful to Logo for giving them an opportunity to clarify the above issues.

Generalization of Hemp's century old theory to multiple loads

Hemp's (1904) exact truss optimization theory has not been extended to stress-based multi-load trusses until recently, when Rozvany, Sokol and Pomezanski (2014) filled this significant gap in our knowledge. Their results will be briefly reviewed in the lecture.

Concluding remarks

It will be seen that the field of structural topology optimization has not been free of controversies, but these have been useful in clarifying certain misconceptions in the literature.

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Implementation of Modern Design of Experiment (MDOE) on Wind Tunnel Plunging Tests of Standard Dynamics Model (SDM)

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Abstract
As the wind tunnel tests (dynamically and statically tests) are very important part of design of every aerial vehicles and also has much cost, design of experiment in this area would be very useful and necessary. The conventional test approaches, include One Factor at a Time (OFAT), have some problems, such as cost, time and don't evaluation of interaction between tests variables in aerodynamics, stability and control aspects of design of aerial vehicles and etc.. OFAT means that if we have for example 3 test variables, we should change one variable and the other variables should be constant. This article has written to implement a process on wind tunnel tests for reduce the number of tests and also evaluate the interaction of design factors (tests variables). This process, which is named MDOE, has some steps. First, factors and domain of each factors and also limitation of wind tunnel tests of standard dynamics model should be specified. Secondly, test points are designed based on the Design of Experiments (DOE) methods. In this article we use full factorial method for specification of test points. Finally, a RSM fitted based on the wind tunnel results in DOE points. The RSM method which is used in this research is second order polynomial. After the fitting of RSM, error of each RSM should be calculated. This research implements MDOE on the SDM in wind tunnel test and each RSM are evaluated by four factors of error evaluation. This research have shown that MDOE has a good and space filling response in entire of design space in particular case.
Keywords: Wind Tunnel, Modern Design of Experiments (MDOE), Standard Dynamics Model (SDM), Response Surface Methodology (RSM), Response Surface Methodology error.

Introduction

All wind tunnels should do research and innovate in some areas such as wind tunnel testing mechanisms and facilities, design of test matrix for each test procedure and teach researchers. It is important that all sections of wind tunnels be updated. Design of test matrix is one of the most important section of each wind tunnel testing.

Design of Experiments (DOE, of which Response Surface Method (RSM) is a subset) have historically targeted engineers and scientists. In recent years, the aerospace community has begun formulating methods to exploit the benefits of DOE with regards to vehicle wind tunnel testing [1], [2]. The DOE approach differs from the OFAT approach because it is process oriented rather than task oriented. DOE methods approach an experiment by identifying all design factors (independent variables) and all desired response (outputs).

Modern Design of Experiments (MDOE)

Wind tunnel tests, are included of measurement forces and moments, in statically and also dynamically states. For implementing the MDOE to reduce the tests number and increase the accuracy, the test designer should know the limits of test factors (test variables). In addition, with respect to limits and type of tests the method of Design of Experiment (DOE) should be known. After the design of experiment method, the test

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designer should test the model in wind tunnel with respect to the DOE points. After the tests, RSMs should be prepared. The mentionable point is that MDOE unlike OFAT procedure can evaluate the interaction of test variables. All DOE methods have some positive and negative points which the experimenter should choose one of them based on his test type and expected results.

First step: Determination and evaluation of the test target and limitation of wind tunnel, type of test and its obligations.

Second Step: Determination of test matrix based on the first step.

Third Step: Choose one of the DOE methods for the test procedure. After the determination of DOE methods, wind tunnel test should be done. In this research, 3^k Full Factorial Design (FFD) is applied for this step.

Forth Step: Create RSM based on the wind tunnel test results. There are some methods for creating the RSM. In this research, the polynomial (second order) method is applied.

Fifth Step: After the creating RSM, the models should be evaluated and verified. If the models be accurate, MDOE process is finished and the model introduce as the response of the wind tunnel test, otherwise there is a loop to create new DOE points and do other steps again.

Wind Tunnel Test Procedure

The experiments were conducted in a trisonic wind tunnel. It is a continuous open circuit tunnel with test section dimensions of $60 \times 60 \times 120$ cm. The test section Mach numbers vary from 0.4 to 2.2 via the engine RPM and different nozzle settings. All oscillatory data were taken at Mach numbers of 0.4. Corresponding to the Reynolds numbers of 0.84e7 per meter respectively. For the plunging motion, the static angles of attack were 0, 6 and 12 degrees and the plunging amplitudes were ± 1, ± 3 and ± 5 cm with the same oscillation frequencies as those of the pitching motion, i.e. 1.25, 2.77 Hz.

The model considered in the present experiments was typical of a fighter aircraft called the standard dynamics model (SDM) and has been used in many research centers for flowfield study and verification of dynamic test rigs for several years [2-4]. It has 32 cm length and 10.34 cm semi span. Figure 1 shows this model.

Verification of RSMs

The error and accuracy of the RSMs can be evaluated by 4 factors which are arranged based on the results and responses.

$$R^2 = 1 - \frac{\sum_{i=1}^{n} (y_i - Y_i)^2}{\sum_{i=1}^{n} (y_i - y_{bari})^2} \tag{1}$$

$$\delta_{bar} = (\frac{1}{n})\sum_{i=1}^{n} \delta_i, \quad \delta_i = | y_i - Y_i | \tag{2}$$

$$\sigma_\delta = \sqrt{\frac{\sum_{i=1}^{n} (\delta_{bar} - \delta_i)^2}{n-1}} \tag{3}$$

$$RMS = \sqrt{\frac{\sum_{i=1}^{n} (\delta_i)^2}{n}} \tag{4}$$

Which Y_i is result of RSM, y_i is the result of wind tunnel, n is the number of test and y_{bari} is the average of the all wind tunnel results in test points. The accuracy standard of R^2 is proximity to 1 and for other error factors are to proximity to 0. It should be mentioned that these factors that are introduced for physical and computational experiments, are totally different from residual error [4]. In this research all of these factors are evaluated for second and third order RSMs.

Results and Discussion

The main subject of this article was implementation of MDOE on plunging wind tunnel tests of SDM. This article is shown that it is possible which a high cost and longtime testing procedure of wind tunnels have been reduced and with an appropriate method have an accurate response in entire of test space.

It is shown in results that 3^k FFD is a good DOE method for wind tunnel tests because it is good space filling and also require low number test points.

Another point is the coordination of error and accuracy factors to calculation of the accuracy. As the tables show, all error and accuracy factors are coordinated and this show which it is possible to use each of them as error and accuracy factor.

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Pictures

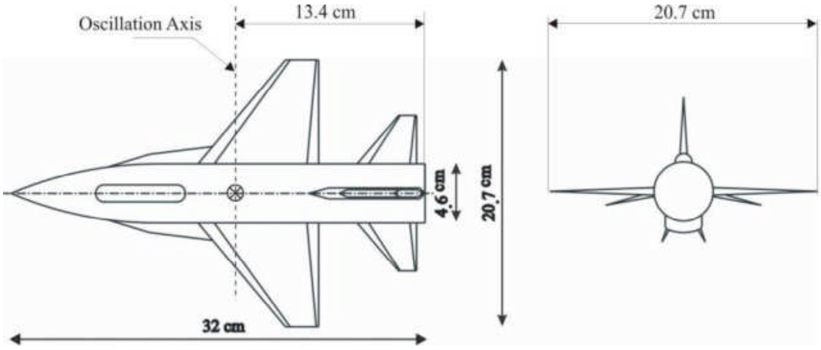

Fig 1. Standard Dynamic Model (SDM).

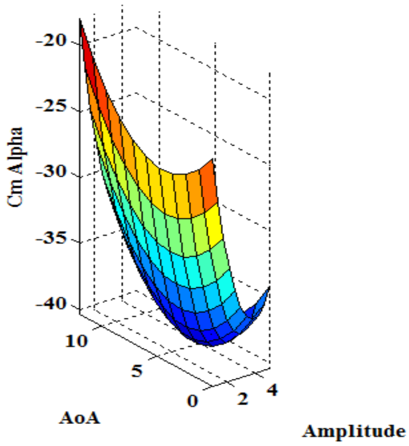

Fig 2. C_{m_α} **in M=0.4, fr=2.77 HZ.**

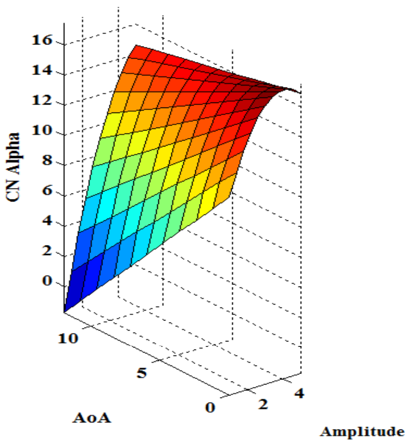

Fig 3. C_{N_α} **in M=0.4, fr=1.22 Hz.**

Optimisation of a MW scale offshore vertical axis wind turbine

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Renewable energy is central to the UK government's objectives to reduce carbon dioxide emissions by 30% by 2020 and to generate 15% of the UK's electricity supply from renewable sources by 2020. It will support economic growth in the UK and increase security of energy supply. With onshore wind farms already making a considerable contribution in the UK, the key opportunities for larger scale development going forward lie offshore. The UK has the world's most ambitious plans to develop offshore wind. However, current progress towards this goal has been accompanied by a significant increase in the capital costs for off-shore wind – largely associated with capacity issues. Furthermore, conventional horizontal axis wind turbines (HAWTs) have a number of limitations for offshore operations, particularly in deep water (i.e. over 50m). For example; scalability restrictions, the necessity for high lift installations offshore requiring specialist vessels, high gravitational and aerodynamic moments on the support structure and a need to maintain rotary equipment at heights typically over 60-80m. Conversely, vertical axis wind turbines (VAWTs) have several inherent attributes that offer some advantages for offshore operations, particularly their scalability and low over-turning moments with better accessibility to drivetrain components.

Figure 1: Aerogenerator concept (Picture courtesy of Windpower Ltd / Grimshaws © 2010)

This paper describes the aerodynamic optimisation of a novel floating 10MW VAWT rotor shape offering a low-stress design to minimise manufacturing and maintenance costs of the whole turbine assembly including the supporting structure and foundations. The Aerogenerator conceptual design study, commissioned by the UK Energy Technologies Institute, combined a V-shaped rotor with outer blades that are inwardly inclined to minimise aerodynamic over-turning moments. This paper describes the shape optimisation of a 10MW Aerogenerator V-VAWT rotor. The need to maximise torque and to minimise over-turning moments leads to conflicting design requirements so a numerical optimisation procedure was developed to obtain a compromise between aerodynamic efficiency and mechanical and structural constraints for the bearing and support structure. The design studies proposed a 'sycamore' shaped rotor, illustrated in figure 1, as a credible alternative to current offshore wind turbine designs and concepts [1]. A non-linear, gradient-search type, constrained optimisation routine was used in the optimisation procedure. The optimisation routine was coupled to an aerodynamic performance model developed for the project based on the Double-Multiple Streamtube (DMST) model [2].

The Aerogenerator design was not necessarily aimed at maximising aerodynamic efficiency but to deliver a low-stress design to minimise manufacturing and maintenance costs. Since the blades of a VAWT rotor see an inconsistent angle of attack through its rotation, they generally use symmetrical aerofoils with a lower lift-to-drag ratio than cambered aerofoils tailored to maximise horizontal axis wind turbine rotor performance. A further design consideration therefore was the feasibility of circulation controlled (CC) VAWT blades, using a tangential air jet to provide lift and therefore power augmentation. However CC blade sections require a higher trailing-edge thickness than conventional sections giving rise to additional base drag. The choice of design parameters is a compromise between lift augmentation, additional base drag as well as the power required to pump the air jet [3].

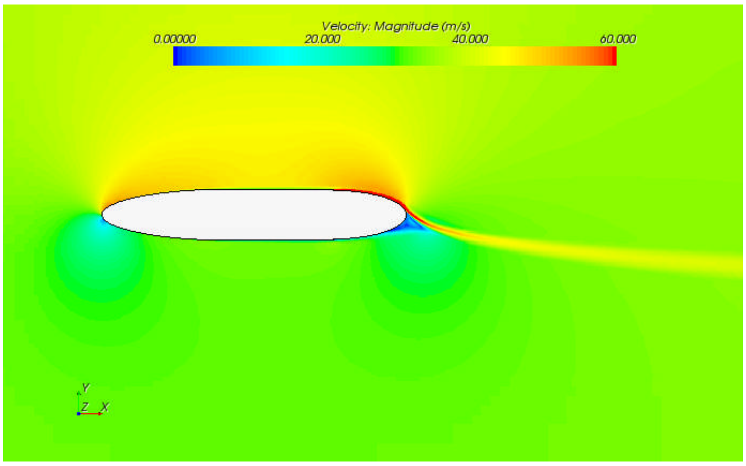

Figure **2**: Predicted velocity contours for a CC aerofoil, AoA = 0^{o}

Computational Fluid Dynamics (CFD) was initially used to derive performance trend data for different CC aerofoil shapes and blowing momentum coefficients as illustrated in figure 2. A numerical optimisation routine was then employed, again coupled to the DMST rotor performance method, to determine an optimum combination of trailing-edge radius, blowing momentum coefficient and nozzle height. The augmented power was offset by the power required to pump the air jet in order to derive a practical solution. The study demonstrated that for modest momentum coefficients significant net power augmentation can be achieved using simple elliptical trailing edge shapes if blowing is controlled through the blades rotation.

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Isogeometric Shape Optimization for Quasi-Static Mechanical Problems

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Abstract

The development of isogeometric analysis (IGA) has triggered renewed interest in shape optimization due to the seamless integration between computer aided design and analysis [1-3]. Traditionally, shape optimization problems have been mostly limited to static loads. In the present contribution, the formulation of shape optimization is extended to include time-dependent quasi-static loads and responses. A general objective functional is used to accommodate both structural optimization and passive control for mechanical problems. An adjoint sensitivity analysis is performed at the continuous level [4,5] and subsequently discretized within the context of IGA.

The methodology is illustrated by considering problems where an external load is allowed to change as a function of time. A first example pertaining to structural optimization is shown in Fig. 1, where a plate with an orifice is subjected to a load that varies continuously from axial compression to simple shear. The objective in this case corresponds to minimizing the difference between the local and the average von Mises stress. As shown in Fig. 2, the optimal shape is influenced by the loading process from compression to shear.

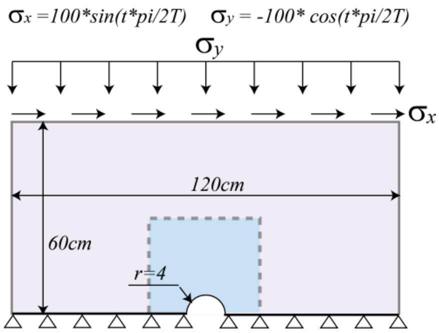

Fig. 1: Plate with an orifice under compression and shear loads changing with time

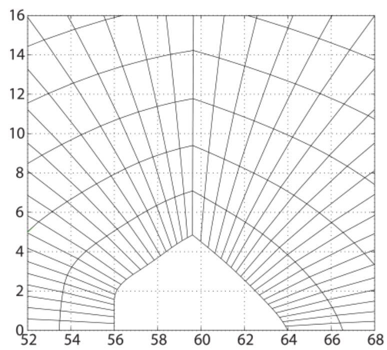

Fig. 2: Optimal shape of the orifice

A second example, which illustrates a passive control formulation, corresponds to finding a shape of a structure such that the point where a moving load is applied remains on a predefined path. In particular, a beam-like structure is subjected to a vertical load that moves along the upper surface, as shown in Fig. 3.

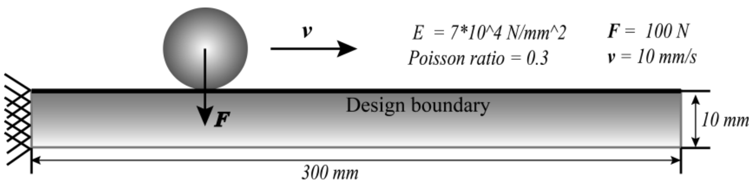

Fig.3: Cantilever beam-like structure under a moving load

The upper surface is given by a function $y = y(x)$, which, upon deformation, displaces to $y + u_y$, where u_y is the vertical displacement. The objective, in this case, is to find the undeformed shape of the structure such that the deformed point where load is applied remains on a curve given by $y = y_0(x)$. This control-like problem is then formulated as

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$$\text{Min G:} = \int_0^T \int_\Gamma (u_y + y - y_0)^2 \delta(x - vt) \mathrm{d}\Gamma \mathrm{d}t$$

$$s.t. \quad \int_\Omega d\Omega \le V_0$$

where V_0 is a given maximum volume of the structure Ω, Γ is the upper surface, v is the velocity of the moving load and T is the total time required for the moving load to move along the upper surface. The optimal shape is shown in Fig. 4 for the special case when $y_0 = 0$, which corresponds to a straight horizontal path. In that example, the original structure is a. Fig. 5 shows the path of the contact loading point for an initial design (a rectangular cantilever beam as shown in Fig.3) and the optimal design shown in Fig. 4. As shown in the Fig. 5, the contact loading point for the original shape corresponds to the classical deflection of a beam loaded at x (i.e., a cubic function of x). In contrast, the contact loading point for the optimized shape remains in a horizontal line (see Fig. 5). The optimal solution is further illustrated in Fig. 6, which shows the deformed shape and the corresponding contact loading point for different times during the loading history.

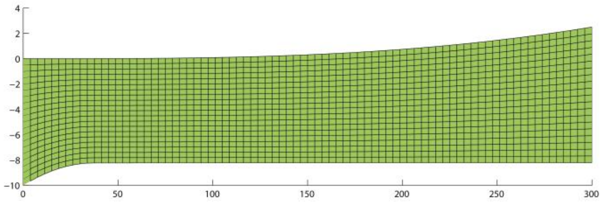

Fig. 4: Optimal shape of the structure

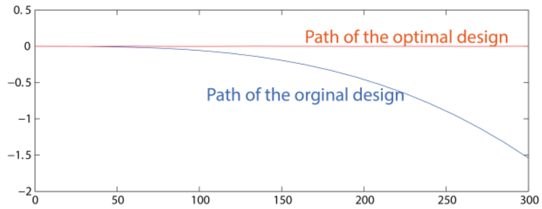

Fig. 5: Comparison between paths of the original and optimal design

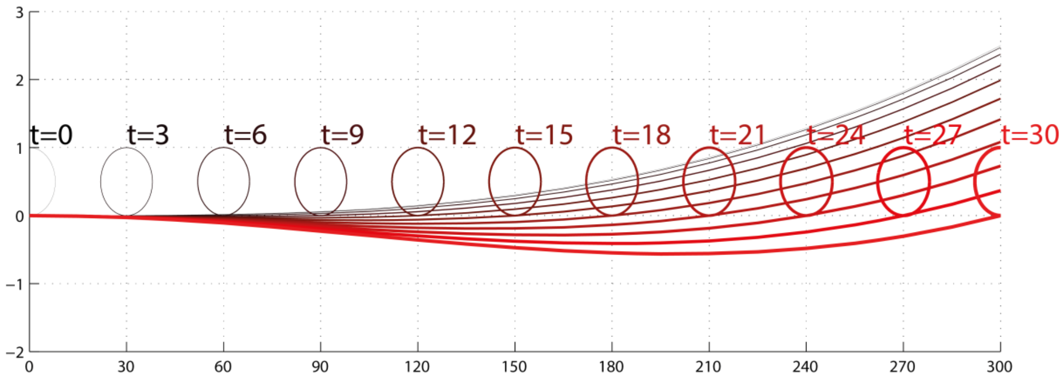

Fig. 6: Load bearing point positions with deformation at different times during the loading history

In general, the examples shown indicate that the methodology developed for quasi-static processes can be employed to systematically solve problems that are relevant for a variety of technological applications in the framework of structural design and inverse problems.

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Specifying MDO coupling structure using the Psi language: a LED lighting design case

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Abstract

In De Borst *et al.* (2013) we analyzed the multidisciplinary design coupling structure of a prototype LED System-in-Package (SiP) lighting device. We used the specification language Psi to define the input-output relationships between design variables, responses, objectives, and constraints. Psi was originally developed as a linguistic software tool for the specification of partitioned problems in decomposition-based design optimization (Tosserams *et al.* 2010a, 2010b). From the Psi specification of the LED SiP design case we automatically generated a design structure matrix, which was subsequently partitioned and sequenced to analyze the coupling structure.

The specification of the multidisciplinary coupling by means of the Psi language presented a clear advantage. For the LED SiP case, the hundreds of variables and responses made it almost impossible to enter the zeros and ones in the matrix by hand. Through the use of the Psi language we avoided the manual entry of the matrix. The effort shifted towards a decomposition-based linguistic specification of the multidisciplinary coupling following a mixed object and aspect decomposition of the LED SiP, and a subsequent assembly of the various components and subsystems into the full SiP description. The Psi language constructs allowing a local specification of variables and response functions for smaller parts of the system and the subsequent manual linkage of the various parts provided the necessary means to effectively model the multidisciplinary coupling in the LED SiP.

However we experienced that the original Psi language was not particularly equipped for the scale of the design problem encountered in the LED SiP design. A rather lengthy Psi specification resulted. From the Psi specification we observed opportunities to enhance the Psi language such that the specification becomes more compact and readable when the scale and complexity of the MDO problem grows.

We present our revision of the Psi language to accommodate for the scale of the LED SiP design problem. In particular, we have added data types and we have changed the syntax regarding the specification of the linkage of variables. A new compiler has been developed, which generates as output a directed bipartite graph, from which for instance the adjacency matrix representation can be obtained.

We demonstrate the Psi language revision by means of a simplified LED SiP example problem, and report what we have gained for the industry-scale LED SiP design problem.

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