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Topology Optimization- Problem Formulation and Pragmatic Outcomes by integration of TOSCA and CAE tools

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***Abstract*—Structural optimization tools have grasped enormous applications in engineering design and development activities because of increasing demands of lightweight and rigid products. In this milieu, researchers have exploited the potentials of nonparametric structural optimization tools like topology optimization by coupling and integrating with other compatible softwares. The first part of this paper deals with the mathematical physical fundamentals of the problem formulation of topology optimization. In the second part application of topology optimization is described by adopting an integrated design approach. The objective is to find the optimal design proposal of a thing walled aerospace component, for which a predecessor design subsists. ANSYS is used to develop the FE model of initial design space of component. TOSCA is used for topology optimization, data reduction and smoothing of optimization results for CAD compatible output. The optimized model is then imported in CATIA to incorporate necessary refinements for manufacturing, machining and geometric restrictions. To validate this new design proposal, optimized model is then imported in ANSYS and analyzed again under the stipulated loads and boundary conditions. At the end, a comparison is made between the predecessor design and the new optimized design proposal. Comparison validates the new design proposal assisted by optimization and simulations is more reliable and reduced weight with enhanced structural performance.**

***Index Terms*— Non parametric optimization, Topology Optimization, TOSCA, CAE, Product design**

I. INTRODUCTION

Currently, impetus of all the modern engineering enterprises have been concentrated to develop reliable, efficient and reduced weight products within least possible development times. To cope with these multitudes of requirements, companies are now enormously depending and have adopted structural optimization tools, like topology optimization in their design and development activities, due to obvious advantages. Topology optimization is used in the concept and predesign phase of the component design process. It reduces product development times by generating an optimal load compatible initial layout design. The innovative design proposal is achieved by removing material in form of holes from a maximum available delineated design space under the specified loads and constraints [1]. During the entire process location of nodes remain unchanged, whereas the material distribution is modified in each optimization iteration. Advanced topology optimization can be executed with different combinations of objective functions and constraints, multiple load cases and desired manufacturing restrictions [2].

Since last decade, many spurs research efforts have been carried out in context of mathematical physical foundation of topology optimization [3, 4, 5]. Modern topology optimization algorithms are outcomes of these research efforts. Besides this, tantamount strives by different academic and commercial organizations to develop the structural optimization softwares have made possible the effective use of topology optimization as a design tool. Presently, many structural optimization softwares have been developed and interfaced with other CAE softwares. For example, OptiStruct by Altair and TOSCA by FE Design are more common, while other analysis softwares have dedicated modules or features of topology optimization like MSC.Nastran, ABAQUS, ANSYS, etc.

Today, applications of topology optimization are very diverse, almost in every field of engineering, from nano tech to aircrafts. For instance, in automobile, Audi is applying topology optimization as an effective tool in the design and development activities [6, 7]. Particularly, in aircraft industry, topology optimization is being applied valiantly, typical examples are redesign of A380 leading edge ribs [8], fuselage wing ribs [9,10], EADS A400M Military Transport Aircraft [11] etc.

Currently, results of topology optimization have become pragmatic due to smoothing and data reduction features embedded in the optimization softwares. These features enable a CAD compatible output, for example in TOSCA software this step is accomplished by TOSCA.smooth [12].

II. MATHEMATICAL PROBLEM FORMULATION OF TOPOLOGY OPTIMIZATION (MINIMUM COMPLIANCE DESIGN)

Nomenclature:

Ω^{mat}	Domain occupied by the body
Ω	Large reference domain
Γ	Lipschitz boundary
f	Body force

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t	Boundary tractions on the traction part $\Gamma_T \subset \Gamma \equiv \delta_\Omega$
Γ_u	Boundary where displacements are prescribed
Γ_t	Boundary where tractions are prescribed
R^n	Metric space, where n = 2,3
E_{ijkl}	Elasticity tensor of order 4
$E_{ijkl}(x)$	optimal stiffness tensor
E_{ad}	Set of admissible stiffness tensors
K	Stiffness matrix
E_e	Stiffness of element for example e = 1, 2, 3…, N
K_e	Global level element stiffness matrix
U	Kinematically admissible displacement fields
E^o_{ijkl}	Stiffness matrix of a given isotropic material

The problem of topology optimization is typically regarded as material distribution problem for the optimal shape design or designing for minimum compliance or maximum global stiffness [1]. Problem formulation is based on basic equations of elasticity and some variational and energy principles. Mathematically, problem can be expressed by considering a boundary value problem for an elastic body subjected to external forces that is volume forces (f), surface tractions (p), and concentrated forces (F_k) [13]. The elastic body confines a domain having volume V and surface S. This is shown in figure 1.

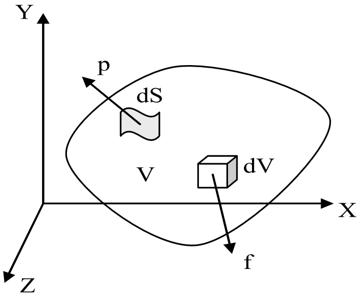


Fig. 1 Elastic body subjected to body and surface forces

Body occupies a subdomain Ω^{mat}, which is a part of larger reference domain Ω in three dimension (R3). This is shown in figure 2. The outer surface is sum of the part of the boundary with displacements u (Γ_u), part of the boundary with tractions t that is (Γ_t) and the free surfaces where tractions are zero.

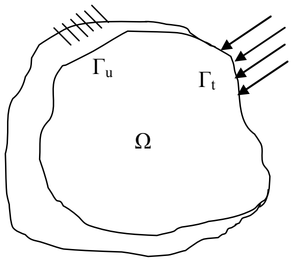


Fig. 2 Three dimensional domain subjected to surface forces

Optimal design is concerned with the problem of finding the optimal choice of stiffness tensor E_{ijkl} with reference to Ω. Now assume the body undergoes a virtual displacement δu from its equilibrium state. The virtual work of an elastic body, due to external forces and virtual displacements is defined as [14]:

$$\delta W = \int_V f\delta u\, dV + \int_S p\delta u dS + F\delta u^0$$

In vector notation;

$$\delta W = \int_V f^T \delta u\, dV + \int_S p^T \delta u dS + F^T \delta u^0 \quad (1)$$

Where,

f^T	Vector of volume forces
p^T	Vector of surface tractions
F^T	Vector of concentrated forces $(F1^T, F2^T \ldots\ldots Fn^{T)}$
$(u^0)^T$	Vector of displacement vectors for concentrated forces
$(u\ )^T$	Displacement vector (u, v, w)

Assume that the elastic body is subjected to a virtual displacement then the virtual work done due to external forces is [15]

$$\delta W = \int_V F\delta u\, dV + \int_S p^T \delta u dS \quad (2)$$

For the mixed boundary value problems, surface tractions can be expressed as;

$$p^T = (\sigma_{i_j} nj)St$$

So (2) can be expressed more elaborately:

$$\delta W = \int_V F\delta u\, dV + \int_S \sigma_{ij} n_j \delta u dS \quad (3)$$

Because, the virtual displacements are vanished on S_u, so the integral domain of the first integral can be changed to S. By means of this, surface integral can be changed to a volume integral. Therefore (3) can be transformed into:

$$\delta W = \int_V F\delta u\, dV + \int_V \sigma_{ij} n_j \delta u\, dV$$

$$\delta W = \int_V F_i \delta u_i dV + \int_V (\sigma_{ij}\delta u_i)_{,j}\, dV$$

$$\delta W = \int_V F_i \delta u_i\, dV + \int_V (\sigma_{ij}\delta u_i)_{,j}\, dV$$

$$\delta W = \int_V F_i \delta u_i\, dV + \int_V (\sigma_{ij}\delta u_i)_{,j}\, dV$$

$$\delta W = \int_V F_i \delta u_i\, dV + \int_V (\sigma_{ij,j}\delta u_i + \sigma_{ij}\delta u_{i,j})dV \quad (4)$$

In scalar notations, equilibrium equations are expressed as:

$$\sigma_{ji,j} + F_i = 0 \quad (5)$$

For example:

$$\frac{\partial \sigma_x}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + F_x = 0$$

And the strain displacement relations are expressed as;

$$\varepsilon_{ij} = \frac{1}{2}(u_{i,j} + u_{j,i}) \quad (6)$$

Therefore from (4), (5) and (6):

$$\delta W = \int_V F_i \delta u_i\, dV + \int_V (-F_i \delta u_i + \sigma_{ij}\varepsilon_{ij})dV$$

$$\delta W = \int_V F_i \delta u_i\, dV + \int_V (\sigma_{ij,j}\delta u_i + \sigma_{ij}\delta u_{i,j})dV$$

$$\delta W = \int_V \sigma_{ij}\varepsilon_{ij} dV \quad (7)$$

The stress-strain relationship with an elasticity tensor of order 4 can be expressed as:

$$\sigma_{ij} = E_{ijkl}\varepsilon_{kl}$$

Therefore, (7) can be transformed into:

$$\delta W = \int_V E_{ijkl}\, \varepsilon_{kl}\, \varepsilon_{ij}\, dV \quad (8)$$

If the equilibrium state is defined as ‘a’ and virtual displacement as ‘b’, then by means of energy bilinear form virtual work done by the elastic body (for a reference domain Ω) can be expressed as [1]:

$$\delta W(a,b) = \int_{\Omega} E_{ijkl}(x)\varepsilon_{ij}(a)\varepsilon_{kl}(b)dV \qquad (9)$$

Where x with $E_{ijkl}(x)$ indicates the optimal stiffness tensor.

$$\delta W(a,b) = \int_{\Omega} E_{ijkl}(x)\varepsilon_{ij}(a)\varepsilon_{kl}(b)dV$$

Load linearized form can be expressed as

$$l(a) = \int_{\Omega} fud\Omega + \int_{\Gamma_T} tudS \qquad (10)$$

Therefore, from above expressions, the minimum compliance or maximum global stiffness problem, with linearized strains and the load linear form is expressed as:

$$\min \ l(a)$$

Subject to

$$\delta W_E(a,b) = l(b) \text{ for all } b \in U$$

$$E \in E_{ad}$$

The above expression is written in its weak variational form. E_{ad} indicates that problem is formulated as a distributed and discrete value design problem. For solving the above problem by computational means we need to discritize the problem through FEM approach. If the structure is discretized into N finite elements, then by using FEM procedures the above expressions take the form;

$$\min \ f^T u$$

Subject to:

$$\min \ K(E_e)u = f$$

$$E_e \in E_{ad}$$

Where K is stiffness matrix which depends on the stiffness of the individual elements E_e,

$$K = \sum_{e=1}^{N} K_e(E_e)$$

Where; e = 1, 2… N and K_e represent the global level element stiffness matrix.

Because, the topology optimization is concerned with determining the optimal selection of material points and voids (nonmaterial areas) of a given design domain, so for the reference domain Ω, problem is to determine the optimal subset Ω^{mat} of material points. For this E_{ad} consists of only those tensors for which the following conditions are satisfied:

$$E_{ijkl} = g(x)\, E^{o}_{ijkl}$$

Where g(x) is a design variable

$$g(x) = \begin{Bmatrix} 1 & if & x \in \Omega^{mat} \\ 0 & if & x \in \Omega / \Omega^{mat} \end{Bmatrix}$$

Subject to:

$$\int_{\Omega} g(x)d\Omega = Vol(\Omega^{mat}) \leq V$$

Where V is the total initial volume before optimization and E^{o}_{ijkl} is the stiffness tensor of given isotropic material.

III.TOPOLOGY OPTIMIZATION
(INTEGRATED DESIGN APPROACH)

In this section, nonparametric topology optimization has been executed and an integrated design approach is implemented to find the optimal design proposal of a thin walled component that is being used in the vertical stabilizer of an aircraft. The objective is to improve the component structure for which a predecessor design subsists. The integrated design approach is illustrated in figure 4. Topology optimization of the component has been accomplished by applying nonparametric structural optimization software TOSCA, in a batch-process mode with ANSYS.

Loads and constraints sought for the existing design have been considered as the baseline for the new design proposal. The first step is to develop a 3D model of maximum available design space, either in CAD or CAE. In this case, model is developed in ANSYS, as shown in figure 3. The model is developed by considering the optimized and nonoptimized regions. These are required to be defined subsequently during the optimization take formulation. After defining the material properties, loads and constraints, model is meshed and solved for load stress analysis.

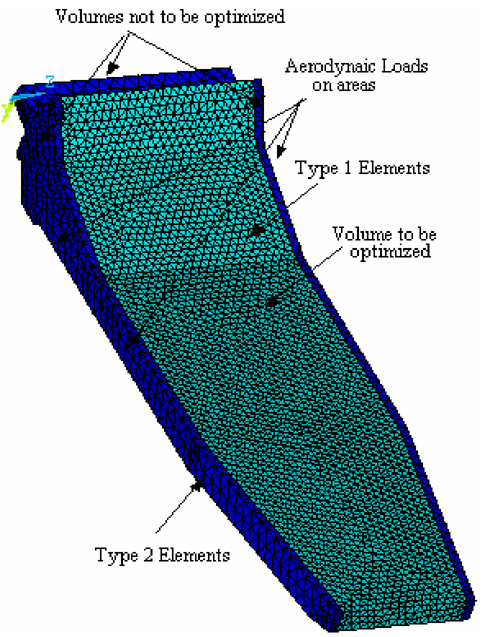


Fig.3. Maximum available design space

Data of the initial 3D model is then archived to a text file, in .cdb format. In the next step, optimization problem is defined in TOSCA.pre module. For this, model.cdb file generated by ANSYS is imported in TOSCA database and then used as a finite element input deck, for example definitions of elements, nodes and material properties. Objective function is taken as minimizing compliance or to maximize the stiffness. Material properties of each finite element are defined as design variables.

Group of elements which need not to be optimized are defined as frozen elements. In this case, all the elements in the expanse of the vertical stabilizer skins and some structural parts attached with the selected component are considered. Some additional manufacturing constraints are

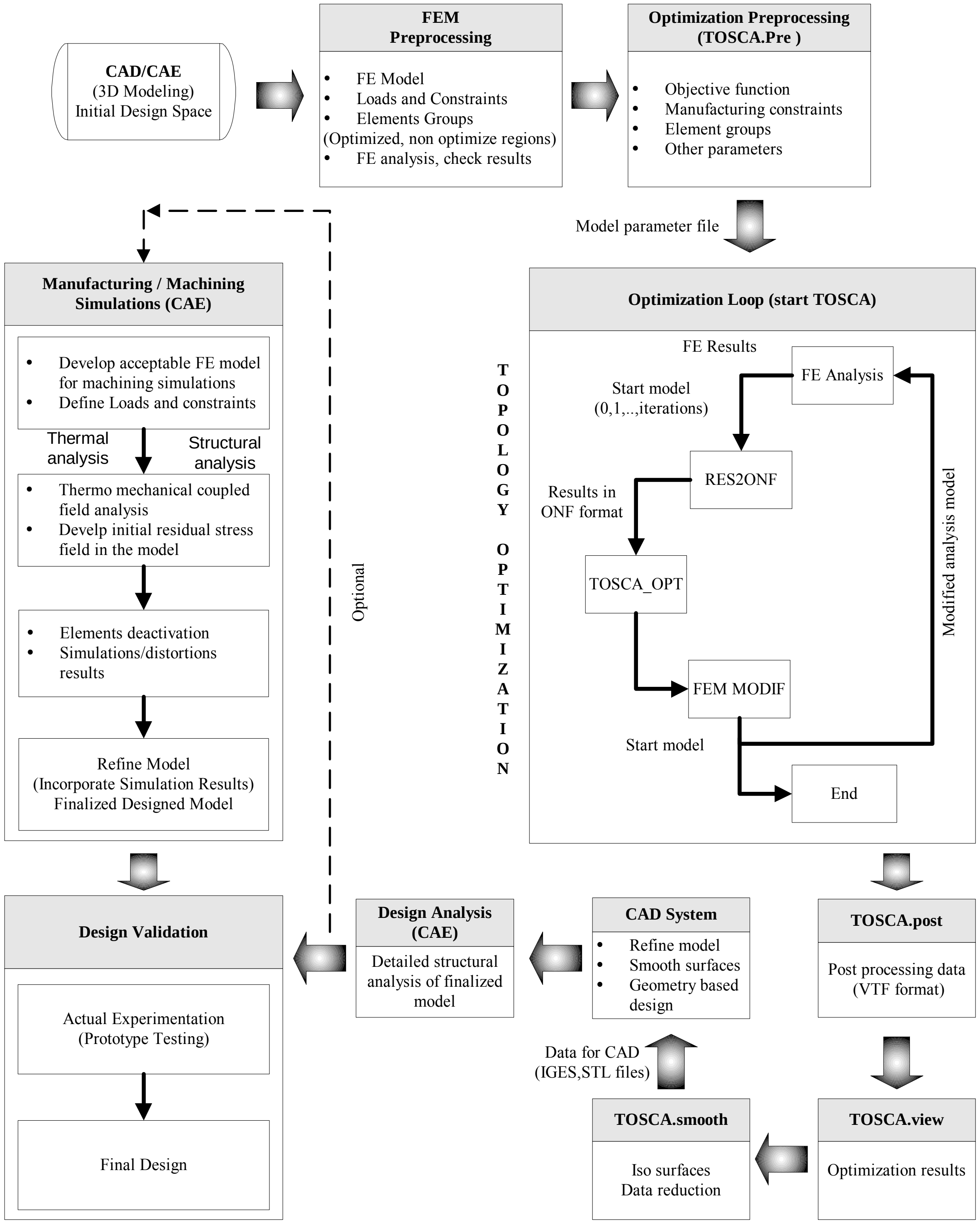
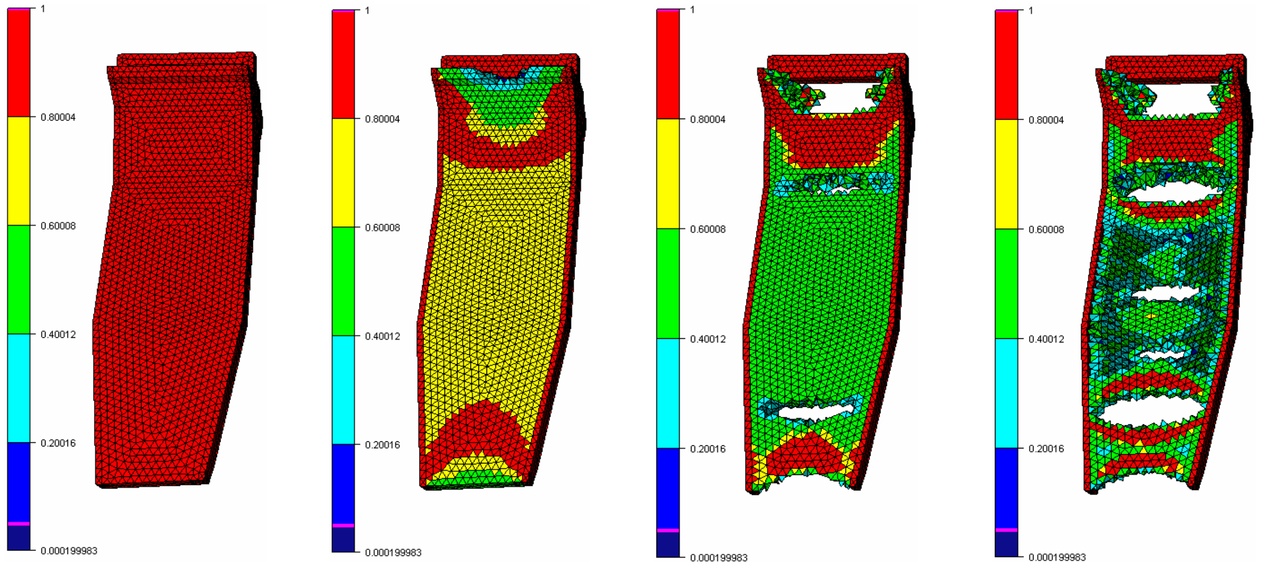


Fig. 4. Integrated design with TOSCA and CAE

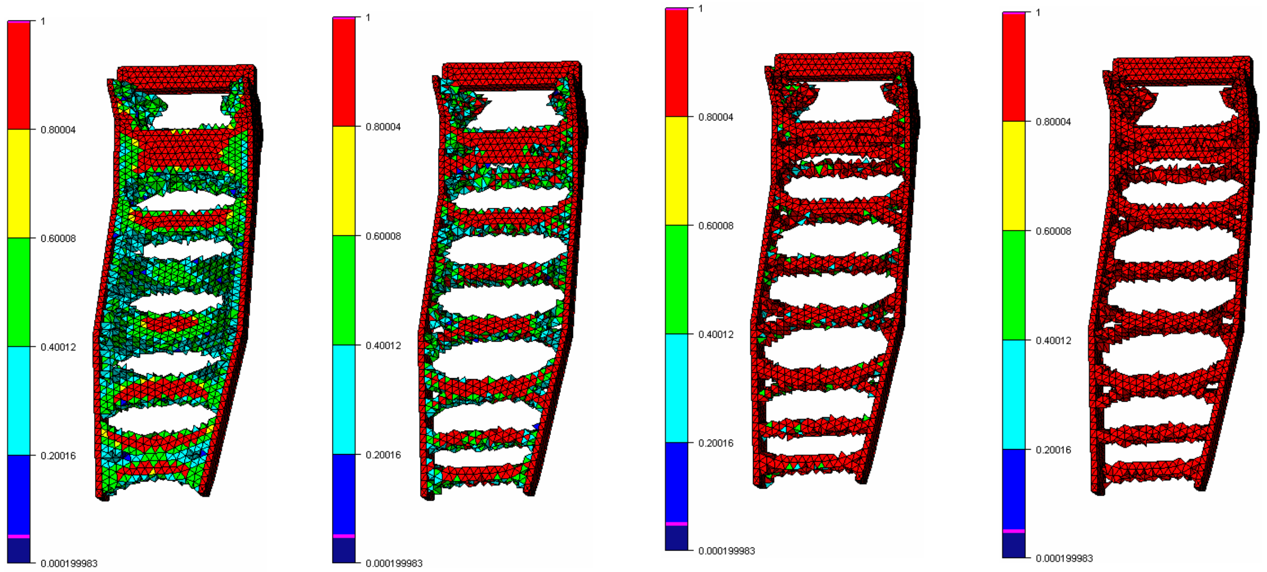
obvious to be defined in most of the cases for viable design solutions, because these constraints directly affect the design variables. The target volume is defined as the 30% of the initial design space. The TOSCA input commands are saved in TOSCA database as a model parameters file in model.par format.

The next step is executing the posed problem. This is performed by the Start.TOSCA module with ANSYS solver. During the each optimization iteration, FE model and parameter file is launched and solved by the solver simultaneously. Results of each optimization iteration are saved in optimization neutral format (.onf), and material distribution in FE model is modified accordingly.

The optimization cycle terminated successfully after 32 iterations. TOSCA.post has been used to transform the (.onf) files format to (.vtf) format. Subsequently, optimization results have been visualized by the TOSCA.view module. The results are shown in figure 5. Tosca.smooth is used for data reduction, smoothing and calculations of iso surfaces. Smoothing of the results is essential for interpretation of data as a CAD compatible format (STL,igs or VRML) and for any subsequent simulation analysis. Using the last optimization iteration results and an ISO value of 0.3, ISO surfaces calculated by TOSCA.smooth are shown in figure 6.



Optimization Step 0 Optimization Step 1 Optimization Step 7 Optimization Step 14



Optimization Step 17 Optimization Step 28 Optimization Step 35 Optimization Step 35

Fig. 5. Topology optimization results – Material distribution

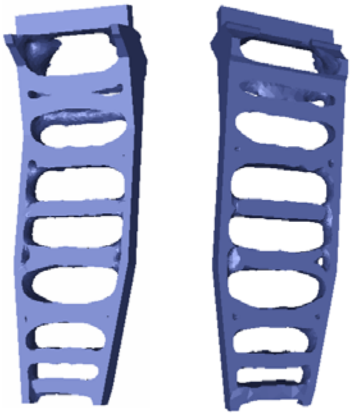


Fig. 6. Isosurfaces calculated by TOSCA.smooth

An IGES (model.igs) file created by TOSCA.smooth is then imported in CATIA, as shown in figures 6 and 7. The optimized results are then further refined in CATIA, by accounting for the manufacturing and functional constraints that could not be accomplished by the whole optimization loop, for example, machining limitations like minimum size holes, sharp edges etc. Machining viability of the refined model is carried out by CATIA virtual machining module. Structural performance of the finalized model is checked by executing the load stability and stress analysis under the same loading and constraints as previously specified on initial design space. The last step is to validate and finalize the new design proposal with actual experimentation and prototype testing.

As thin walled structures are prone to distortions and dimensional instability due to pre and post machining induced stresses [16], therefore cutting simulation analysis via CAE can be performed to have more robust design proposal. In figure 3 this is shown as an optional step in integrated design approach. Through these simulations and analysis, a designer can deduce post machining situations, and can induce appropriate manipulations at the design stage to avoid distortions and dimensional inaccuracies that generally deemed to be inevitable. Cutting simulations also aid in determining optimized cutting sequences. The actual machined component that is being processed by the aircraft company is shown in figures 8 and 9.

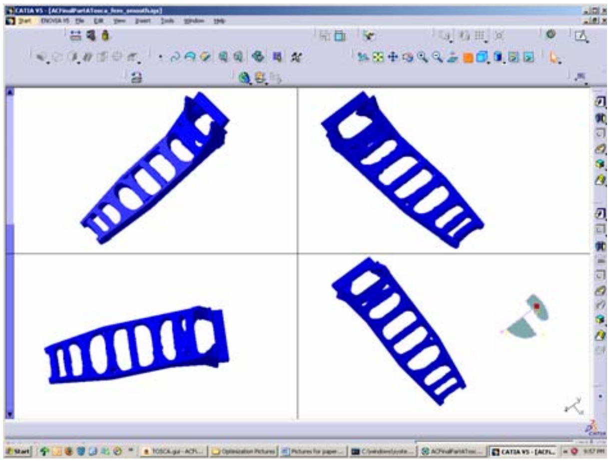


Fig. 7. Model imported in CATIA

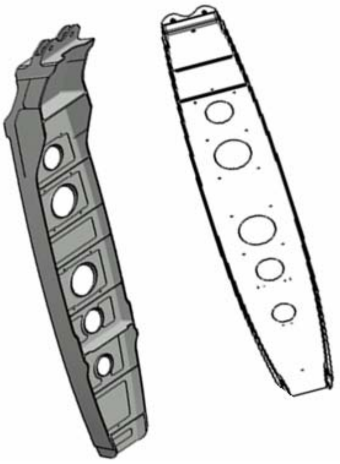


Fig.8. Existing design (Aircraft vertical fin component)

Fig. 9. Actual machined component

IV. CONCLUSIONS

In this paper a brief overview of mathematical foundations and pragmatic solutions of topology optimization approached with nonparametric structural optimization tools have been presented. An integrated design environment with nonparametric optimization, CAE and CAD system aids in finding the optimal design proposals. However it largely depends on the accuracy to formulate the optimization problem, interfaces and compatibility of data formats. TOSCA optimization modules with data smoothing feature provide a comprehensive approach to find an optimal design proposal at the early design phase.

Topology optimization can be used for dual purposes, to improve the existing designs and to find the novel design proposals. For instance, if a comparison is made between the predecessor design and the new design proposal, evidently the new design is better in all aspects. For example, it is lighter in weight, have better structural performance under the same loading conditions and easier to machine. In predecessor design weight is reduced by designing the multipockets on front end while at back end, there is no material within side thin walls. In the new design proposal these have been modified with through pockets and integrally machined stiffeners within outer thin walls. Concurrently, topology optimization is an important tool to find the optimal design proposals with least possible times, because it reduces the overall product development cycle time.

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