OPTIMAL DESIGN OF ACTIVELY-COOLED PANELS STRENGTHENED WITH VARIOUS CELLULAR MATERIALS*

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This article focuses on the thermal-structural optimal design of actively-cooled panels reinforced by low density cellular structures. Three types of core topologies are considered, i.e., hexagonal honeycombs, lattice trusses in pyramidal configuration, and corrugated sheets. Parametric numerical models of actively-cooled panels are built up to perform the thermal-structural analysis. The thermal-structural analysis is implemented considering thermal load and internal pressure from the combustion gases, and thermal load and internal pressure from the coolant, while the scramjet combustor is working at Mach 6. Then, an optimization procedure is carried out to find the lightest design while satisfying thermal deformation and plastic strain constraints, with the thickness of plates and size of cellular structures as design variables. The results demonstrate that, compared with traditional actively-cooled panels, the weight reduction for actively-cooled panels reinforced by cellular structures is significant, and the optimal design may reach a weight reduction as high as 33.4%.

Keywords: *thermal-structural analysis, actively-cooled, sandwich panel, light-weight, optimization*

1 Introduction

Components that experience extreme heat flux, while simultaneously supporting mechanical loads, are frequently encountered in hypersonic aircrafts. The challenge is more severe for a scramjet combustor. It is an effective measure to use active cooling structures in a scramjet to guarantee a long duration engine operation. The key problem is how to design lighter actively-cooled structures with good performance in thermal insulation and load bearing.

Metallic sandwich structures have been investigated for their lightweight and multifunctional characteristics, such as thermal insulation, shock resistance, and vibration suppression. The mechanical benefits of sandwich structures with various topologies have been well-documented in the recent literature [1-3]. Metallic sandwich structures with periodic truss and prismatic cores have been processed and used to construct lightweight and compact heat sinks because of the potential for simultaneous load bearing and active cooling[4,5]. However, there are still some challenges for this actively-cooled structures in consideration of their practicability, mainly due to large pressure drop and the need of a great deal of coolant liquid. Structural optimization of sandwich panels has been addressed [2,6,7], however, little work has been done on the complex structure of a combustor sandwich panel that subject to complex thermal-mechanical loads.

In this paper, we proposed a class of actively-cooled panels reinforced by cellular structures, which combines merits of traditional cooling channels in heat transfer and merits of sandwich structures in sustaining mechanical load. Parametric numerical models are built up to perform the thermal-structural analysis when a combustor is working at Mach 6. Based on the sub problem approximation method, the optimization of actively-cooled panels reinforced by sandwich structures is implemented.

2 Structure of Actively-cooled Panel

For actively-cooled panels, a variety of shapes can be envisioned for cooling ducts. The present study focuses on rectangular ducts (Fig.1). The section dimensions of rectangular ducts and the space between the ducts of are fixed at $\delta 1$=1.5mm , δ=3mm respectively

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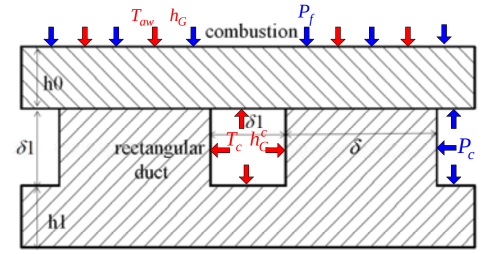

Figure 1: Schematic of actively-cooled panel, with a representative unit cell

The cores illustrated in Fig.2 represent prototypical topologies of current interest. One is the example of truss-based core, in pyramidal configuration. The other two are based on plate elements in prismatic configurations; specifically, hexagonal honeycomb and corrugated sheet. In general, these cores geometry is characterized by three parameters: core member thickness, t, core thickness h ,and core member length, c. But for the pyramidal configuration, the angle ω between the core members and the face sheets is indentified. The length of the panel is 788mm, and the width is 35mm.

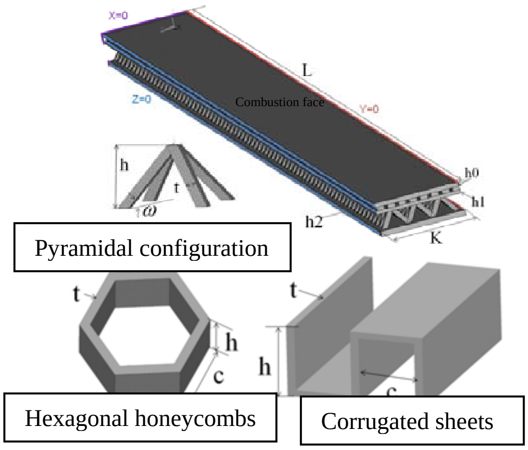

Figure 2: Schematics of three prototypical core topologies and reinforced panel

3. Thermal-structural Analysis

The sequentially coupled physics analysis is adopted to study the thermal-structural characteristics of the actively-cooled panels reinforced by sandwich structures. Steady-state thermal analysis was firstly implemented to determine temperature distributions. Convective heat transfer boundary conditions at Mach 6 [8] are applied both to the combustor face and the internal channel surfaces. The temperature variations along the panel length have been taken into consideration.

$$\begin{aligned} -k(T)\frac{\partial T}{\partial n}\Big|_{\Gamma_1} &= h_G(T_{aw}-T_{w1}) \\ -k(T)\frac{\partial T}{\partial n}\Big|_{\Gamma_2} &= h_G^c(T_c-T_f) \end{aligned} \quad (1)$$

The principle of finite element analysis for a thermal-structure problem is shown as follows.

$$KU = P \quad (2)$$

where K is stiffness matrix, U is displacement vector, and P is load vector.

$$K^e = \int B^T DBdv \quad (3)$$

where K^e is element stiffness matrix, B is strain-displacement matrix, and D is stress-strain matrix.

$$P^e = \int B^T D\alpha(T\text{-}T_0)dv \quad (4)$$

where T is element temperature, T_0 is reference temperature, and α is expansion coefficient vector

Then, the element stress can be obtained:

$$\sigma = D[BU\text{-}\alpha(T\text{-}T_{0.})] \quad (5)$$

Here, a bi-linear elastic-plastic model for 304 stainless steel, which is the material used in the panels, is adopted in the analysis, and the nonlinear variation of thermal parameters and mechanical properties with temperature is considered.

A prototypical combustor wall is subject to three loading mechanisms: external pressure (P_f =5MPa) from the combustion gases, internal pressure (P_c=1.5MPa) from the coolant and thermal loads due to the temperature differences between the combustion side and the vehicle exterior.

The mechanical boundary conditions are considered as shown in Fig 1. Uniform thermal expansion is permitted in all directions. The external pressure can cause panel-level bending and the internal pressure can bend individual face segments.

4. Optimization Procedure

The ANSYS program uses two optimization methods to accommodate a wide range of optimization problems. One is sub problem approximation method that can be efficiently applied to most engineering problems. The other is first order method which is based on design sensitivities and is more suitable for problems requiring high accuracy.

The sub problem method is adopted for its practicability and reliability. This method requires only the values of the dependent variables (objective function and state variables). The dependent variables are first replaced with approximations by means of least squares fitting, and the constrained minimization problem is converted to an unconstrained problem using penalty functions. Minimization is then performed every iteration on the approximated, penalized function until convergence is achieved or termination is indicated. Since the method relies on approximation of the objective function and each state variable, a certain amount of data in the form of design sets is needed. This preliminary data can be directly generated by the user using any of the other optimization tools or methods. If not defined, the method itself will generate design sets at random.

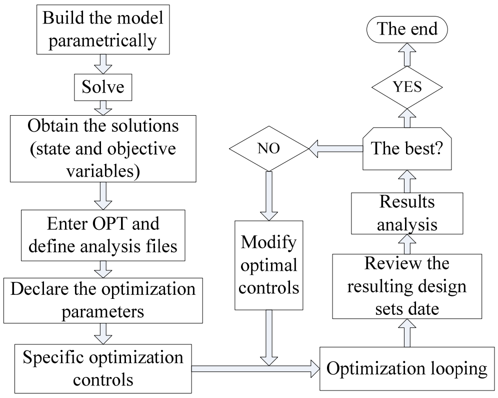

Figure 3: ANSYS optimization process

The flowchart of optimization process is illustrated in Fig.3. For the optimization in ANSYS, the program performs a series of analysis-evaluation-modification cycles. That is, an analysis of the initial design is performed; the results are evaluated against specified criteria, and the design is modified as necessary. The process is repeated until all specified criteria are met.

Declaration of the optimization parameters is a very important step. The design variables (DV), state variables (SV) and objective function (OBJ) in the present optimization problems are considered as the following:

Design Variables (DV):

——Four-parameter optimization is performed for actively-cooled structure reinforced by pyramid sandwich panels: three sheets thickness, h0, h1, h2, and core member thickness, t.

——For the other two structures, a fifth parameter can be identified: the height of the cores, h .

——To keep the optimization tractable, some parameters are fixed (ω=45^o , h=7.07mm for pyramidal configuration, c=6.40mm for hexagonal honeycomb, and c=7.75mm for corrugated sheet).

State Variables (SV):

The optimal design is limited by vertical displacement ($u_y \le 1mm$) and von Mises plastic strain ($s_p \le 0.02$) of the actively-cooled panels.

Objective Function (OBJ):

The objective of the optimization is to find the geometric parameters that minimize weight. Since the same material is used in the optimization, volume of the structure is used as the objective function instead of weight.

4. Results and Discussions

The iteration process of optimal parameters is plotted in Fig 5, including design variables and objective function. For the sub problem method adopted here, the optimizer initially generates random designs to establish the state variable and objective function approximations. After these random designs, the solution terminates quickly when convergence is reached.

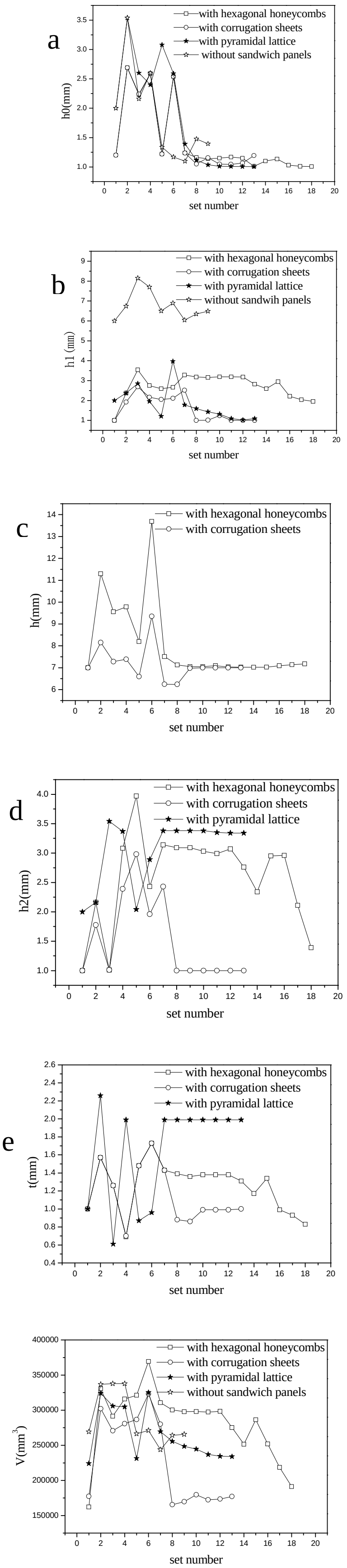

Figure 5: Optimized curve of variables (a)-(e) are design variables (f) is objective function

Table 1: Lightweight rate of different structures

Structure	V(mm^3)	Lightweight rate η
SJ(1)	263750	0
SJ(2)	177100	33.4%
SJ(3)	233980	12.0%
SJ(4)	191170	28.4%

*SJ (1)-without sandwich panels
SJ (2)-with corrugation sheets
SJ (3)-with pyramidal lattice
SJ (4)-with hexagonal honeycombs

$$\eta = \frac{SJ(i) - SJ(1)}{SJ(1)} \qquad i = 2,3,4 \tag{6}$$

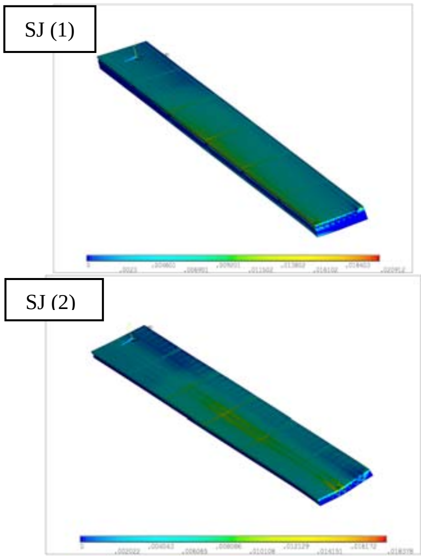

Figure 6: Comparison of von Mises plastic strain distributions

An assessment of performance of the optimized actively-cooled structures is made through comparisons of their weights. The results are presented in Table 1. The weight rank of the various sandwich panels, in decreasing order, is: actively-cooled structure without sandwich panels, with pyramid sandwich panels, with hexagonal honeycomb sandwich panels, and with corrugation sandwich panels. The weight reduction for actively-cooled panels reinforced by sandwich structures over traditional actively-cooled panels is evident.

For the pressure loads and extreme heat flux, plastic strain may occur in the most highly stressed locations for actively-cooled panels. Although the temperature differences, and hence the thermal stresses, are greatest at the channel inlet, the material strength is also greatest at this location. Typical strength reductions with increasing temperature suggest the possibility of the maximum plastic strain at the outlet, where the temperature is at its maximum. For the core channel shape, the internal fuel pressure may induce large stresses. The external pressure may cause large vertical displacement and stresses. Representative von Mises plastic strain distributions of two actively-cooled panels are presented in Fig.6. The plastic strain distributions are different, especially for highly strained location. The sandwich structures may be the main reason for this discrepancy.

5 Conclusions

A thermal-structural analysis and optimization design for lightweight actively-cooled panels reinforced by sandwich structures has been described and implemented. The results indicate that the optimal design method is efficient. The combination of traditional actively-cooled panel with sandwich structure may reach a weight reduction as high as 33.4%, while maintaining the thermal protection capability and load sustaining capability. It may be a new way to design strong and compact heat sinks.

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