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Design Considerations for Thermally Insulating Structural Sandwich Panels for Hypersonic Vehicles

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Abstract

Simplified thermal/structural sizing equations were derived for the in-plane loading of a thermally insulating structural sandwich panel. Equations were developed for the strain in the inner and outer face sheets of a sandwich subjected to uniaxial mechanical loads and differences in face sheet temperatures. Simple equations describing situations with no viable solution were developed. Key design parameters, material properties, and design principles are identified. A numerical example illustrates using the equations for a preliminary feasibility assessment of various material combinations and an initial sizing for minimum mass of a sandwich panel.

1 Introduction

Thermal protection systems (TPS) are essential to protect hypersonic and atmospheric entry vehicles from severe aerodynamic heating. A variety of approaches to thermal protection have been proposed over the years – many of which can be categorized as shown in Fig. 1. and described in detail in Ref. [1]. For acreage areas on reusable vehicles, passive concepts (the top row in Fig. 1) are generally preferred. Heat sink structures are limited to very brief heat pulses, and hot structures are limited by available materials and difficult interfaces with internal components of the vehicle. Therefore, the primary emphasis for reusable vehicles has been on insulated structural concepts such as the aluminum structure protected by the reusable ceramic tiles and blankets of the Space Shuttle Orbiter. Although the tiles and blankets work well as thermal insulators, they form a fragile, high maintenance exterior surface for the vehicle. Optimizing an exterior insulation system for low mass will likely result in a system that is susceptible to surface damage from hazards such as handling, tool drop, launch debris, high speed flight through rain, and hypervelocity impacts in orbit. One way to increase durability is to add mass to the outer surface of the insulation. In addition to the vehicle performance reduction from the added mass, another difficulty is that, depending on the TPS concept, it may be difficult to achieve the required level of durability.

The addition of mass to the vehicle outer surface for improved durability raises the intriguing possibility of using that mass to help carry structural loads. This deceptively simple idea will be difficult to achieve because it requires the outer surface to act as a flight weight aerospace vehicle skin that not only carries the required mechanical loads, but also accommodates severe transient heating with the corresponding hot outer surface. If the outer surface does not carry all of the structural skin loads (then it would be a hot structure), it must have a structural connection to and be compatible with the inner structural skin.

One of the primary functions of an insulated structure TPS is to limit the maximum temperature of the structural skin of the vehicle, so that efficient structural materials can be used and for a simpler interface to the vehicle interior. If a continuous outer skin is connected to an inner structural skin with limited maximum temperature capability, then some means of limiting the heat transfer from the outer hot skin to the cooler inner skin is needed. A sandwich panel with a thermally in-

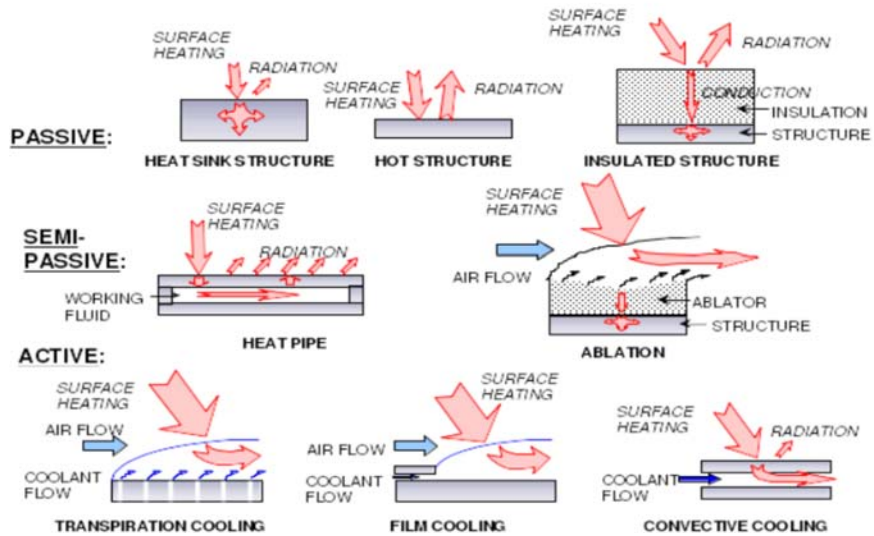

Figure 1. Approaches to thermal protection

sulating, structural core has the potential to provide both the structural connection and the thermal insulation required between the face sheets.

Thermally insulating structural sandwich panels have been studied sporadically over the years. The original multiwall [2] concept was conceived as a multilayer metallic wall construction designed to contain cryogenic fuel, carry vehicle structural loads, and act as a TPS. Another early example was a proposed superalloy honeycomb sandwich structure that was also intended to contain cryogenic fuel, carry vehicle structural loads, and act as a TPS (Refs. [3], [4], and [5]). Slots were cut in the outer face sheet to alleviate some of the predicted thermal stresses. NASA sponsored university research investigated aspects of design and analysis of structurally integrated TPS (Refs. [6], [7], and [8]). Concurrently, a multi-center NASA project focussed on building a prototype structurally integrated TPS and led to an edgewise compression test of a panel [9]. Most previous efforts have focussed on point designs and specific fabrication efforts. One study (Ref. [10]) investigated a simplified analytical model of the thermal performance of a thermally insulating structural panel to identify key design drivers and a simplified sizing methodology. This paper seeks to develop similar insight into the structural performance of a thermally insulating structural panel.

In this paper, a simplified combination of in-plane mechanical load and thermal stress was investigated in an attempt to gain basic insight that will be required to develop optimum sandwich panels that can simultaneously insulate and carry structural loads. Equations were derived to size the sandwich face sheets, so that they do not exceed their allowable strains. Equations were also derived to describe situations where no feasible solution exists (i.e., one or the other face sheet will exceed its strain limit.) Several numerical examples are presented and key design drivers are identified.

2 Structural Design Considerations

TPS and structural sandwich skins of aerospace vehicles each have a multitude of requirements and design objectives that must be considered to develop a viable design. Combining both functions into a single element further complicates the process. The myriad of important details in the design process tend to obscure the key design drivers. The challenge is to simplify the problem for clarity in understanding without losing the essence of it.

The basic challenges of designing a TPS have been understood for quite a while [11]. More recently, the design challenges and objectives for reusable metallic TPS were thoroughly discussed in Chapter 5 of Ref. [1]. Although applied to metallic TPS, the issues are similar for any reusable TPS. The primary function of the TPS is to protect the interior of the vehicle from excessive aerodynamic heating throughout the design lifetime of the TPS. The TPS must also maintain an acceptable aerodynamic surface that does not result in augmented heating that could damage the vehicle. Minimizing mass is always critical. Gaps and seams can require as much or more development effort than the basic TPS itself. Depending on the vehicle mission, other design objectives such as surface durability, low maintenance requirements, quick turnaround, and all weather operation may be important.

There are also many challenges to designing a structural skin for an aerospace vehicle. First, the materials must not exceed their design stress or strain limits when subjected to all the design load cases. The structural panels must not buckle under compressive or shear loads. For sandwich panels, there are additional localized failure mechanisms such as face sheet wrinkling, face sheet dimpling, and various core failure modes that depend on the core configuration. A thorough discussion of issues involved in designing and making structural sandwich panels is given in Ref. [12]. Panel to panel joints and connections to underlying structure also significantly complicate the design.

The design of both TPS and sandwich structures have been widely studied and are fairly well understood, so what are the unique challenges that arise when combining both functions into a single element? 1) The sandwich core must act as a thermal insulator in addition to performing the usual structural functions. 2) The sandwich face sheets will experience large changes in temperature during a vehicle mission and the two face sheets will be at greatly different temperatures at different times.

The thermally insulating function of the core was investigated in Ref. [10]. The materials and configuration of a thermally insulating core that can also carry the required loads will be highly dependent on the design thermal heating history and the design mechanical loads. Because of the wide variety of potential configurations and combinations of materials, it would be quite a challenge to develop generic guidelines for developing a thermally insulating structural sandwich core. One candidate core configuration was developed and thermally tested in Ref. [13].

The primary issue with the sandwich face sheets being different materials at different temperatures is that they will expand different amounts. This difference in thermal growth will lead to thermal stresses and possibly out-of-plane deformations. If the panel edges are connected to adjacent panels, so that the bending stiffness is maintained through the joint, the panel will behave as part of a shell and edge rotations will be restrained. The panel will then tend to develop in-plane thermal stresses rather than out-of-plane deformations. If the connections to additional structure are ignored (they may vary widely for each specific design), the face sheets are tied together and must grow to the same length.

A simple problem can be defined, that retains the primary function of the TPS and the most basic structural function while neglecting many important, but complicating TPS and structural requirements and objectives. A thermally insulating structural sandwich is defined with a core sized to supply the required thermal insulation and face sheets sized to carry uniaxial mechanical loads and accommodate thermal stresses from the mismatch in thermal growth between the face sheets. This will result in a lower bound for the sandwich mass. Additional design considerations, such as local and global buckling will likely increase panel mass.

3 Sizing Face Sheets for Uniaxial Loads

The simplified problem investigated in this paper is illustrated in Fig. 2. A sandwich panel with an outer face sheet of thickness, d_f, an inner face sheet of thickness, d_s, and a thermally insulating core of thickness, d_e is subjected to uniaxial loading, N_x with face sheet temperatures of T_f and T_s. The face sheets are assumed to carry all of the load and the core load carrying capability is neglected. Contributions of the core to thermal stress is also neglected. The sandwich is assumed to initially be at a temperature, T_i, at which there are no thermal stresses. Relevant thermal and structural properties of the face sheets and core are listed in Fig. 2.

3.1 Mechanical Loading

The stiffness of the sandwich core is neglected, so the face sheets must carry the entire mechanical load.

$$N_x = d_s\sigma_s + d_f\sigma_f \tag{1}$$

If the panel is not allowed to rotate at the edges and it is loaded through its neutral axis, the panel will remain flat and the strains in the two face sheets will be the same. (Ignore the possibility of buckling for now.) The simple uniaxial stress-strain relations are

$$\begin{aligned} \sigma_s &= E_s\epsilon \\ \sigma_f &= E_f\epsilon \end{aligned} \tag{2}$$

Combining Eqns. 1 and 2 produces

$$N_x = (d_sE_s + d_fE_f)\epsilon \tag{3}$$

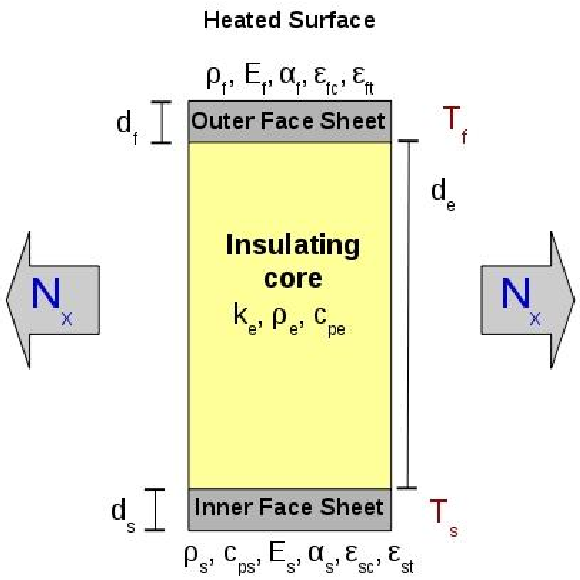

Figure 2. Illustration of a simplified problem

Eq. 3 can be rewritten in terms of face sheet mass per unit area.

$$N_x = (m_s \frac{E_s}{\rho_s} + m_f \frac{E_f}{\rho_f})\epsilon \tag{4}$$

3.2 Thermal Stress

Thermal stresses can be very complex in a built-up structure with transient, spatially varying heat loads. A greatly simplified situation is considered to gain insight into the fundamental behavior of a sandwich panel subjected to heating on one surface. The two face sheets of the panel are comprised of different materials and may be at different uniform temperatures at any instant of time. Inplane stiffness of the sandwich core is assumed to be negligible. The sandwich is assumed to be free of thermal stress at an initial temperature of T_i. The effects of connecting the panel to underlying structure are neglected. Panel edges are assumed to be rotationally restrained, but allowed to grow in plane. The edges of the panel remain perpendicular to the inplane direction, so that the two face sheets are constrained to change length by the same amount. This means that the overall vehicle shell may grow, but individual panels do not locally bend out of plane. If only uniaxial stresses are considered, this problem reduces to the classic "two bar" problem described in Ref. [14]. At some instant of time each face sheet has experienced the following temperature change.

$$\begin{aligned}\Delta T_s &= T_s - T_i \\ \Delta T_f &= T_f - T_i\end{aligned} \tag{5}$$

The thermal stress in the outer face sheet can be found by writing the “two bar” equation from Ref. [14] using the nomenclature of the current paper.

$$\sigma_{fT} = -\frac{E_f(\alpha_f \Delta T_f - \alpha_s \Delta T_s)}{1 + \frac{d_f E_f}{d_s E_s}} \tag{6}$$

Assuming a simple, uniaxial situation, the strain associated with thermal stress for the outer face sheet can be expressed as follows.

$$\epsilon_{fT} = \frac{\sigma_{fT}}{E_f} \tag{7}$$

Combining Eqns. 6 and 7 and writing in terms of face sheet masses gives the following equation for the strain associated with thermal stress in the outer face sheet.

$$\epsilon_{fT} = -\frac{\alpha_f \Delta T_f - \alpha_s \Delta T_s}{1 + \frac{m_f \rho_s E_f}{m_s E_s \rho_f}} \tag{8}$$

Following a similar procedure for the inner face sheet produces the following equation for the thermal stress.

$$\sigma_{sT} = \frac{d_f}{d_s}\left(\frac{E_f(\alpha_f \Delta T_f - \alpha_s \Delta T_s)}{1 + \frac{d_f E_f}{d_s E_s}}\right) \tag{9}$$

The equation for strain associated with thermal stress in the inner face sheet can be written as follows.

$$\epsilon_{sT} = \frac{\alpha_f \Delta T_f - \alpha_s \Delta T_s}{1 + \frac{m_s E_s \rho_f}{m_f \rho_s E_f}} \tag{10}$$

3.3 Combined Mechanical and Thermal Loads

The face sheet materials are assumed to be linearly elastic, so the mechanical strains and the strains associated with thermal stress can be simply added together for a combined load case. For each face sheet, the combined strains must remain within the acceptable tensile and compressive strain limits. These strain limits will depend on the material properties and the design approach (i.e., factor of safety, etc.), which may vary with vehicle and mission.

The combined strain for the outer face sheet can be expressed as

$$\epsilon_{fc} \le \epsilon + \epsilon_{fT} \le \epsilon_{ft} \tag{11}$$

Combining Eqns. 4, 8, and 11 produces the following expression for the outer face sheet strain.

$$\epsilon_{fc} \le \frac{N_x - m_s \frac{E_s}{\rho_s}(\alpha_f \Delta T_f - \alpha_s \Delta T_s)}{m_s \frac{E_s}{\rho_s} + m_f \frac{E_f}{\rho_f}} \le \epsilon_{ft} \tag{12}$$

The combined strain for the inner face sheet can be expressed as

$$\epsilon_{sc} \leq \epsilon + \epsilon_{sT} \leq \epsilon_{st} \tag{13}$$

Similarly, combining Eqns. 4, 10, and 13 produces the following expression for the inner face sheet strain.

$$\epsilon_{sc} \leq \frac{N_x + m_f \frac{E_f}{\rho_f}(\alpha_f \Delta T_f - \alpha_s \Delta T_s)}{m_s \frac{E_s}{\rho_s} + m_f \frac{E_f}{\rho_f}} \leq \epsilon_{st} \tag{14}$$

If a combination of face sheet masses can be found that meets the strain limit constraints of Eqns. 12 and 14 for a set of design loads and temperatures, the corresponding core mass can be found using the methodology of Ref. [10] and the following equation.

$$m_e = \left(\frac{(\frac{\rho_e k_e}{\sqrt{C_{pe}}})^2 {t_h}^2}{2c_{ps}\left(-ln(1-\frac{T_m}{T_h})\right)^2} \right)^{\frac{1}{3}} {m_s}^{-\frac{1}{3}} \tag{15}$$

The mass of the panel is simply the sum of the face sheet masses and the core mass.

4 Cases with No Viable Solution

Inspection of Eqns. 12 and 14 raises the possibility that there may be situations where no combination of face sheet masses can satisfy the strain constraints. For this simple uniaxial mechanical loading, it is always possible to add mass to reduce the resulting strain. Temperature differences put one face sheet in tension and the other in compression, so adding mass to decrease the magnitude of the strain in one will increase the magnitude of the strain in the other. Therefore, the thermal loading has the potential to result in a situation with no viable solution.

For a typical atmospheric entry flight, there are two situations that would likely generate the worst case thermal stresses. The first case would occur when the outer face sheet reaches its maximum temperature and the inner face sheet has not yet risen from its initial temperature. The second case would occur near or after landing when the inner face sheet reaches its maximum temperature, but the outer face sheet has cooled to ambient temperature.

$$\begin{array}{llll} \text{Case 1:} & \Delta T_f = (\Delta T_f)_{max} & \Delta T_s = 0 & N_x = 0 \\ \text{Case 2:} & \Delta T_f = 0 & \Delta T_s = (\Delta T_s)_{max} & N_x = 0 \end{array} \tag{16}$$

For Case 1, the outer face sheet will be in compression and the inner face sheet will be in tension. For the outer face sheet in compression, Eq. 12 simplifies to

$$\epsilon_{fc} \leq \frac{-m_s \frac{E_s}{\rho_s} \alpha_f (\Delta T_f)_{max}}{m_s \frac{E_s}{\rho_s} + m_f \frac{E_f}{\rho_f}} \tag{17}$$

Solving for m_f while recalling that compression strain limits are by definition negative quantities and all other parameters in the equation are positive results in the following expression.

$$m_f \geq -\frac{E_s}{\rho_s}\frac{\rho_f}{E_f}\left(1+\frac{\alpha_f(\Delta T_f)_{max}}{\epsilon_{fc}}\right)m_s \tag{18}$$

For Case 1, the inner face sheet is in tension, so Eq. 14 simplifies to

$$\frac{m_f\frac{E_f}{\rho_f}\alpha_f(\Delta T_f)_{max}}{m_s\frac{E_s}{\rho_s}+m_f\frac{E_f}{\rho_f}} \leq \epsilon_{st} \tag{19}$$

Solving for m_f produces

$$m_f \leq \frac{\frac{E_s}{\rho_s}\frac{\rho_f}{E_f}m_s}{\frac{\alpha_f(\Delta T_f)_{max}}{\epsilon_{st}}-1} \tag{20}$$

Inspection of Eq. 18 reveals that if $|\epsilon_{fc}| > \alpha_f(\Delta T_f)_{max}$, the right hand side becomes negative. The physical significance is that there is enough compressive strain range in the outer face sheet to accommodate the entire thermal expansion mismatch, so the outer face sheet cannot exceed its strain range. However, if $|\epsilon_{fc}| < \alpha_f(\Delta T_f)_{max}$, the right hand side of the equation is positive and it is possible for the outer face sheet to exceed its compressive strain limit. A similar argument can be applied to Eq. 20 for the inner face sheet. Eqns. 18 and 20 are portrayed in Fig. 3 for the case when $\alpha_f(\Delta T_f)_{max}$ is greater than either strain limit. The hatched area represented along the x-axis represents the region where the outer face sheet strain limit would be violated and the hatched area along the y-axis represents the region where the inner face sheet strain limit would be violated. The area with no hatching represents the feasible design space for Case 1. When the slope of Eq. 18 reaches or exceeds the slope of Eq. 20, there is no feasible design space – one or the other design limit will be violated. To mathematically determine when this occurs, Eqns. 18 and 20 can be combined as follows.

$$-\frac{E_s}{\rho_s}\frac{\rho_f}{E_f}\left(1+\frac{\alpha_f(\Delta T_f)_{max}}{\epsilon_{fc}}\right)m_s \geq \frac{\frac{E_s}{\rho_s}\frac{\rho_f}{E_f}m_s}{\frac{\alpha_f(\Delta T_f)_{max}}{\epsilon_{st}}-1} \tag{21}$$

After a few algebraic manipulations the previous equation reduces to this simple expression.

$$\alpha_f(\Delta T_f)_{max} \geq \epsilon_{st}-\epsilon_{fc} \tag{22}$$

This simple equation means that if the thermal growth of the hot outer face sheet exceeds the strain range bounded by the inner face sheet tensile limit and the outer face sheet compressive limit, any combination of face sheet masses will result in the strain limit being exceeded in at least one of the face sheets. Therefore, there is no viable solution.

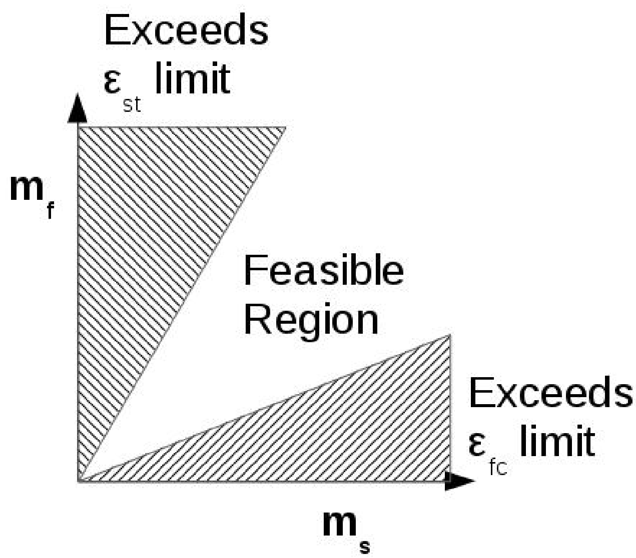

Figure 3. Case 1 feasible design space

For Case 2, the outer face sheet will be in tension and the inner face sheet will be in compression. For the outer face sheet in tension, Eq. 12 simplifies to

$$\frac{m_s \frac{E_s}{\rho_s} \alpha_s (\Delta T_s)_{max}}{m_s \frac{E_s}{\rho_s} + m_f \frac{E_f}{\rho_f}} \leq \epsilon_{ft} \tag{23}$$

Solving for m_f produces

$$m_f \geq \frac{E_s}{\rho_s} \frac{\rho_f}{E_f} \left(\frac{\alpha_f (\Delta T_s)_{max}}{\epsilon_{ft}} - 1 \right) m_s \tag{24}$$

For Case 2, the inner face sheet is in compression, so Eq. 14 simplifies to

$$\epsilon_{sc} \leq \frac{-m_f \frac{E_f}{\rho_f} \alpha_s (\Delta T_s)_{max}}{m_s \frac{E_s}{\rho_s} + m_f \frac{E_f}{\rho_f}} \tag{25}$$

Solving for m_f produces

$$m_f \leq \frac{\frac{E_s}{\rho_s} \frac{\rho_f}{E_f} m_s}{1 - \frac{\alpha_s (\Delta T_s)_{max}}{\epsilon_{sc}}} \tag{26}$$

The same argument used for Case 1 can be applied to Case 2 to define the situation for which there is no feasible design. Eqns. 24 and 26 can be combined as follows.

$$\frac{E_s}{\rho_s} \frac{\rho_f}{E_f} \left(\frac{\alpha_f (\Delta T_s)_{max}}{\epsilon_{ft}} - 1 \right) m_s \geq \frac{\frac{E_s}{\rho_s} \frac{\rho_f}{E_f} m_s}{1 - \frac{\alpha_s (\Delta T_s)_{max}}{\epsilon_{sc}}} \tag{27}$$

The previous equation reduces to this simple expression after a few algebraic manipulations.

$$\alpha_s(\Delta T_s)_{max} \geq \epsilon_{ft} - \epsilon_{sc} \tag{28}$$

If the thermal growth of the warmer inner face sheet exceeds the strain range bounded by the outer face sheet tensile limit and the inner face sheet compressive limit, any combination of face sheet masses will result in the strain limit being exceeded in at least one of the face sheets. Therefore, there is no viable solution. So, if the situation defined by either Eq. 22 or Eq. 28 occurs, then there is no viable solution.

5 Numerical Example

It may be helpful to consider a numerical example to illustrate how the equations derived in this paper can be useful for preliminary design studies. Eqns. 22 and 28 can be rewritten in a form that defines a very useful ratio for preliminary screening of face sheet materials.

$$\frac{\alpha_f(\Delta T_f)_{max}}{\epsilon_{st} - \epsilon_{fc}} \geq 1$$

or

$$\frac{\alpha_s(\Delta T_s)_{max}}{\epsilon_{ft} - \epsilon_{sc}} \geq 1 \tag{29}$$

If either of the thermal strain ratios defined in Eq. 29 is greater than 1, the situation is not viable. If both ratios are much less than 1, thermal stresses are likely not a strong design driver. If either ratio is slightly less than 1, the design may be viable, but thermal stresses will be a major design driver.

5.1 Initial Screening for Face Sheet Materials

The use of the ratios in Eq. 29 for screening combinations of face sheet materials is illustrated by considering an example with four different candidate inner face sheet materials and three candidate materials for the exterior face sheet. The exterior surfaces are subjected to two different surface temperature histories.

The inner face sheet materials are the same four candidates considered in Ref. [10]: aluminum, graphite/epoxy, beryllium/aluminum, and titanium. The thermal properties and temperature limits are the same as those used in Ref. [10], but the structural properties, particularly the strain limits, are somewhat arbitrarily chosen for this example. Property values for the inner face sheet candidates are shown in Table 1.

The outer face sheet material properties used for this example are listed in Table 2. The material labeled CMC1 is typical of a carbon/silicon carbide composite material. Properties for carbon/silicon carbide materials may vary widely from these values. For the other two materials, CMC2 and CMC3, the coefficient of thermal expansion, modulus of elasticity, and strain limits were parametrically varied to decrease the thermal stresses.

Table 1. Material properties for inner face sheet candidates

Material	$T_i + T_m$ $^\circ F$	c_p $\frac{Btu}{lbm^\circ F}$	ρ $\frac{lbm}{in^3}$	α $\frac{in/in}{^\circ F}10^{-6}$	E Msi	ϵ_{sc}	ϵ_{st}	t_{mg} in
Al	300	0.215	0.10	13	10	-0.005	0.005	0.01
Gr/Ep	200	0.218	0.057	2	6	-0.005	0.005	0.02
Be/Al	450	0.404	0.076	9	29	-0.0015	0.0015	0.01
Ti	500	0.137	0.16	5	16	-0.0075	0.0075	0.01

Table 2. Material properties for outer face sheet candidates

Material	ρ $\frac{lbm}{in^3}$	α $\frac{in/in}{^\circ F}10^{-6}$	E Msi	ϵ_{sc}	ϵ_{st}	t_{mg} in
CMC1	0.08	4	30	-0.001	0.0005	0.05
CMC2	0.08	3	25	-0.002	0.00075	0.05
CMC3	0.08	2	20	-0.003	0.001	0.05

Two surface temperature histories are considered for this example. One temperature history is for a location on the Space Shuttle Orbiter during atmospheric entry and the other temperature history is predicted for a location on a single-stage-to-orbit vehicle. Further details on these two temperature histories can be found in Ref. [10]. The maximum surface temperature rise for each temperature history as well as the time, temperature rise and average pressure for an equivalent square heat pulse are listed in Table 3.

Table 3. Two surface temperature histories

History	$(\Delta T_f)_{max}$ $^\circ F$	t_h s	T_h $^\circ F$	P_{avg} psi
BP7490	1758	1273.6	1509	0.014
ATSpA	1784	1914.4	1568	2.55

The ratios defined in Eq. 29 can be easily calculated using the material properties in Tables 1 and 2 along with the temperatures from Table 3. The resulting ratios for each combination of face sheet materials is listed in Table 4 for temperature history BP7490. Calculations for temperature history ATSpA produce nearly identical results because the material properties are the same and there is only a slight difference in the maximum temperature of the outer face sheet.

Ratios greater than 1 in Table 4 indicate that the combination of face sheet materials is infeasible and are shown in bold type. The next to the last column contains ratios for the situation when the outer face sheet is at its maximum temperature and the inner face sheet is still at the initial temperature. The last column to the right contains the ratios for the situation when the inner face sheet has reached its maximum temperature, but the outer face sheet has cooled to ambient temperature. The only inner face sheet material that may work with an outer face sheet of CMC1 is titanium. Beryllium/aluminum, for the properties used in this example, will not work with any of the candidate outer face sheet materials because of its low strain limits and relatively high coefficient of thermal expansion.

Table 4. Thermal strain ratios for temperature history BP7490

Outer f.s.	**Inner f.s**	$\frac{\alpha_f(\Delta T_f)_{max}}{\epsilon_{st}-\epsilon_{fc}}$	$\frac{\alpha_s(\Delta T_s)_{max}}{\epsilon_{ft}-\epsilon_{sc}}$
CMC1	Al	**1.17**	0.57
	Gr/Ep	**1.17**	0.05
	Be/Al	**2.81**	**1.76**
	Ti	0.83	0.28
CMC2	Al	0.75	0.54
	Gr/Ep	0.75	0.05
	Be/Al	**1.51**	**1.56**
	Ti	0.55	0.27
CMC3	Al	0.44	0.52
	Gr/Ep	0.44	0.05
	Be/Al	0.78	**1.40**
	Ti	0.33	0.26

5.2 Preliminary Face Sheet Sizing for Mechanical and Thermal Loading

A thermally insulating structural sandwich panel that acts as the skin of a hypersonic vehicle will be sized for multiple combinations of mechanical and thermal loads. A set of six loads (Table 5) were chosen for this example. The first two loads may represent launch loads, the second two entry loads, and the last two landing loads. However, any number of combinations of inplane mechanical load and face sheet temperatures could have been chosen.

A computer program was written using Version 2.7 of the Python programming language to identify combinations of face sheet masses that do not exceed material strain limits when subjected to the design loads in Table 5. For each face sheet, an array of masses between the minimum gauge mass and 2.0 lbm/ft^2 was generated in increments of 0.02 lbm/ft^2. For each load case in Table 5, the outer face sheet strain (Eq. 12) and the inner face sheet strain (Eq. 14) were calculated for every combination of face sheet masses. The feasible design space is defined as the combinations of face sheet masses for which strain limits were not exceeded in either face sheet for any design load case.

Table 5. Combined mechanical and thermal loads

Load case	N_x $\frac{lb}{in}$	ΔT_f $^\circ F$	ΔT_s $^\circ F$
1	2000	0	0
2	-3500	0	0
3	500	$(\Delta T_f)_{max}$	0
4	-500	$(\Delta T_f)_{max}$	0
5	500	0	$(\Delta T_s)_{max}$
6	-1100	0	$(\Delta T_s)_{max}$

The ratios in Table 4 indicate that the combination of a CMC1 outer face sheet and a titanium inner face sheet may be a feasible design for temperature history BP7490. The feasible design space for a sandwich with this combination of face sheet materials subjected to the loads in Table 5 is illustrated in Fig. 4. The feasible design region is small and the two bounding load cases are constrained by the compressive strain limit in the CMC1 material. The outer face sheet must be considerably thicker than the inner face sheet to avoid exceeding its compressive strain limit. Although there may be a feasible design with this combination of materials, having the outer face sheet so much thicker than the inner face sheet does not seem to be an attractive configuration.

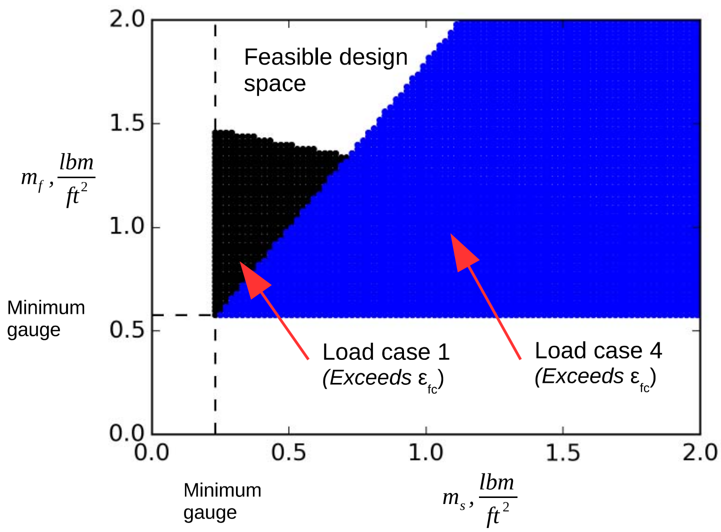

Figure 4. Feasible design space for CMC1 outer face sheet and titanium inner face sheet

The ratios in Table 4 also show that any material except beryllium/aluminum will work for an inner face sheet when combined with CMC2 for the outer face sheet. The feasible design space for a sandwich with CMC2 for the outer face sheet and graphite/epoxy for the inner face sheet, subjected to the loads in Table 5 for the heating history BP7490 is illustrated in Fig. 5. The feasible design region is considerably larger than that shown in Fig. 4. The feasible design space is again partially bounded by the outer face sheet compressive strain limit for load cases 1 and 4. The tensile strain limit of the inner face sheet forms an additional bound to the design space for load case 3. This design space offers the possibility of having a design with the face sheets more nearly the same mass.

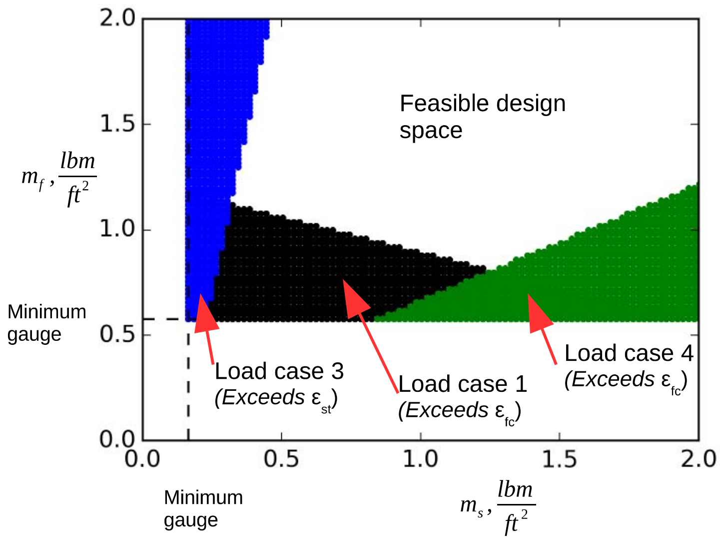

Figure 5. Feasible design space for CMC2 outer face sheet and graphite/epoxy inner face sheet

Although these feasible design spaces have been identified using very simplistic loads and methods, the design space is unlikely to be expanded by considering additional design details. Sizing for buckling, local failure modes, biaxial loading, shear, fatigue and fracture, etc. will probably result in thicker face sheets. However, the combination of face sheet masses must still lie within this feasible design space to avoid exceeding face sheet strain limits.

5.3 Minimum Sandwich Mass

The next step is to determine the best combination of face sheet masses within the feasible design space. Design objectives could include mass, cost, and manufacturing concerns. However, to keep this example simple, the objective is minimum sandwich mass.

The total mass of the face sheets at any point in the feasible design space is simply the sum of the inner and outer face sheet masses. The mass of the thermally insulating core depends only on the mass of the inner face sheet, the thermal properties of the core and the surface temperature and ambient pressure history. The core mass (neglecting structural considerations for the core) can be estimated using Eq. 15. Insulator properties for heating histories BP7490 and ATSpA are listed in Table 11 of Ref. [10]. The mass of a core of AETB-8 required to meet the thermal requirements (structural performance not addressed) for temperature histories BP7490 and ATSpA is shown as a function of inner face sheet mass in Fig. 6. Although the maximum temperatures are nearly the same, the ATSpA is a much longer heating pulse with a correspondingly higher integrated heat load. Therefore, the ATSpA temperature history requires significantly more insulation than the BP7490 temperature history. The higher the inner face sheet mass, the more heat can be stored in the face sheet before the temperature limit is exceeded, so less insulation is required. For the BP 7490 temperature history, the core insulation mass ranges from 4 lbm/ft^2 at minimum gauge to less than 2 lbm/ft^2 at an inner face sheet mass of 2 lbm/ft^2. Similarly, for the ATSpA temperature history the core mass ranges from 7 lbm/ft^2 to about 3 lbm/ft^2, nearly double the required insulation mass.

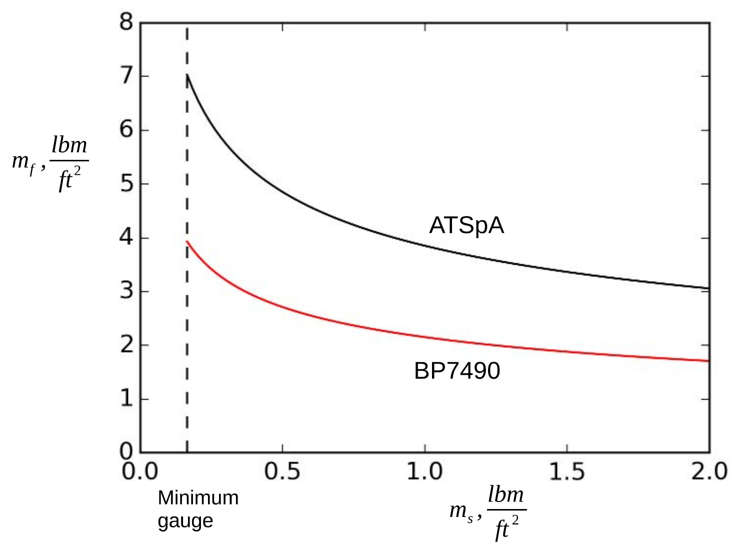

Figure 6. Core mass as a function of inner face sheet mass

A contour plot of the total sandwich mass over the feasible design space of Fig. 5 is shown for the BP7490 temperature history in Fig. 7. The feasible design space is again shown as a function of the two face sheet masses. The colored contour bars span the range from 4 lbm/ft^2 (blue) to 5.75 lbm/ft^2 (red). As expected, the total mass tends to increase as the face sheet masses increase. However, the nonlinear variation of core mass with inner face sheet mass (Fig. 6) causes the sharp curvature of the contour lines for small values of inner face sheet mass. If the core were not included in the total, the contours would be evenly spaced -45 degree diagonal lines. From the contour lines, it is not visually obvious exactly where the minimum mass

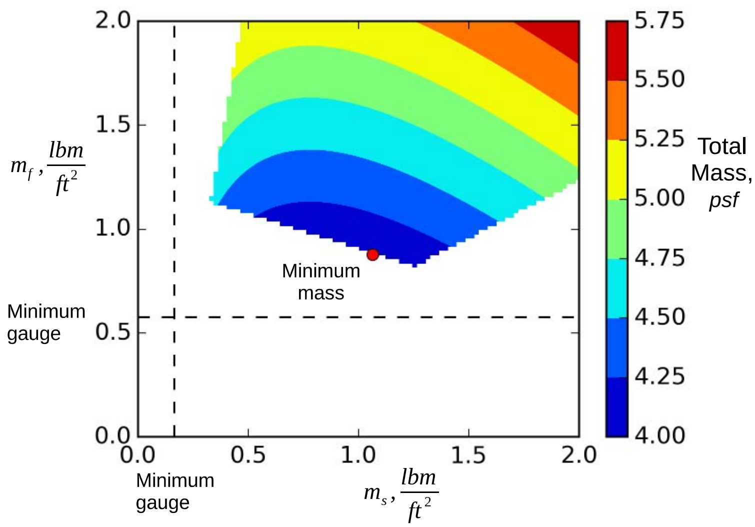

Figure 7. Total mass contours over feasible design space (BP7490)

point is located. The lower boundary is nearly parallel to a portion of the lowest contour line. The red circle indicates the combination of face sheet masses that result in the lowest total sandwich mass. Numerical values for the masses and thicknesses of sandwich components for the minimum mass configuration are given in Table 6.

Table 6. Minimum mass sandwich configuration (BP7490)

Component	**Material**	**Mass** $\frac{lbm}{ft^2}$	**Thickness** in
Outer f.s.	CMC2	0.876	0.076
Inner f.s.	Gr/Ep	1.064	0.130
Core	AETB-8	2.107	3.165
Total		4.047	3.371

A similar contour plot is shown for temperature history ATSpA in Fig. 8. The colored contour bars span the range from 5.5 lbm/ft^2 (blue) to 7.25 lbm/ft^2 (red), significantly higher than the contour range shown in Fig. 7. The shape of the contours is also more significantly affected by the higher core mass for the ATSpA temperature history. In this case, it is obvious that the combination of face sheets for minimum mass will be near the intersection of the boundaries for load case 1 and load case 4 (see Fig. 5).

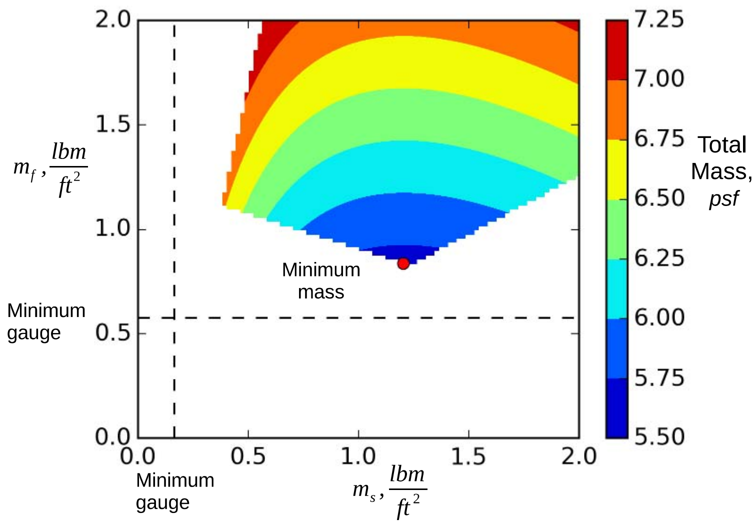

Figure 8. Total mass contours over feasible design space (ATSpA)

Numerical values of the masses and thicknesses of sandwich components for the minimum mass configuration (ATSpA) are given in Table 7. The outer face sheet is slightly thinner and the inner facesheet is slightly thicker than those in Table 6. However, the core is 71 percent thicker because of the much higher integrated heat load. Similar calculations could be performed for the other feasible combinations of

Table 7. Minimum mass sandwich configuration (ATSpA)

Component	**Material**	**Mass** $\frac{lbm}{ft^2}$	**Thickness** in
Outer f.s.	CMC2	0.836	0.073
Inner f.s.	Gr/Ep	1.204	0.147
Core	AETB-8	3.620	5.436
Total		5.660	5.656

face sheet materials identified in Table 4 to identify the configuration with the overall lowest mass or most desirable configuration. However, the calculations presented in this section are sufficient to demonstrate how the equations derived in this paper can be used for the preliminary design of thermally insulating structural sandwich panel.

6 Conclusions

The thermal/structural response of a thermally insulating structural sandwich panel was simplified to a one-dimensional problem in an attempt to identify the most important design drivers and material properties affecting its behavior. Equations were derived for the strains in the sandwich face sheets under combined uniaxial mechanical loads and thermal stresses from differences in face sheet temperatures. The calculated strains can be compared to acceptable tensile and compressive strain limits (determined by the designer) for any combination of inplane mechanical load and face sheet temperatures.

The worst case thermal stress situations for a typical atmospheric entry mission were used to derive simple equations for face sheet material combinations that are not feasible for a given heating history. Simple ratios were defined that can be used for rapid screening of candidate face sheet material combinations. The feasibility of given combination of materials depends on the strain limits and coefficients of thermal expansion of both face sheet materials. They cannot be considered separately.

A numerical example illustrated how to use the equations derived in this paper for preliminary screening of candidate face sheet materials and sizing of a minimum mass thermally insulating structural sandwich panel. Three candidate outer face sheet materials and four candidate inner face sheet materials were considered for a sandwich panel subjected to six load cases for two different heating histories. The combinations of face sheet materials were screened for feasibility. Two feasible combinations of face sheet materials were selected for further investigation. For each of these material combinations, a feasible design space was mapped out for which each combination of face sheet masses did not violate any strain limits for any load case. Contour plots of total sandwich mass over these feasible design spaces were used to identify minimum mass sandwich panel configurations.

The equations and methods described in this paper provide insight and tools that can be helpful for developing viable structural panels that also function as a thermal protection system for a hypersonic vehicle.

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14. ABSTRACT Simplified thermal/structural sizing equations were derived for the in-plane loading of a thermally insulating structural sandwich panel. Equations were developed for the strain in the inner and outer face sheets of a sandwich subjected to uniaxial mechanical loads and differences in face sheet temperatures. Simple equations describing situations with no viable solution were developed. Key design parameters, material properties, and design principles are identified. A numerical example illustrates using the equations for a preliminary feasibility assessment of various material combinations and an initial sizing for minimum mass of a sandwich panel.					
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