Analysis of Stainless Steel Sandwich Panels with a Metal Foam Core for Lightweight Fan Blade Design

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   The quest for cheap, low density and high performance materials in the design of aircraft and rotorcraft engine fan and propeller blades poses immense challenges to the materials and structural design engineers. Traditionally, these components have been fabricated using expensive materials such as light weight titanium alloys, polymeric composite materials and carbon-carbon composites. The present study investigates the use of a sandwich foam fan blade made up of solid face sheets and a metal foam core. The face sheets and the metal foam core material were an aerospace grade precipitation hardened 17-4 PH stainless steel with high strength and high toughness. The stiffness of the sandwich structure is increased by separating the two face sheets by a foam core. The resulting structure possesses a high stiffness while being lighter than a similar solid construction. Since the face sheets carry the applied bending loads, the sandwich architecture is a viable engineering concept. The material properties of 17-4 PH metal foam are reviewed briefly to describe the characteristics of the sandwich structure for a fan blade application. A vibration analysis for natural frequencies and a detailed stress analysis on the 17-4 PH sandwich foam blade design for different combinations of skin thickness and core volume are presented with a comparison to a solid titanium blade.

I. Introduction

TRADITIONAL designs of aircraft and rotorcraft engine fan and propeller blades have been fabricated with light weight titanium alloys, polymeric composite materials and carbon-carbon composites. For example, General Electric's current generation GE90 fan blades consist of polymer matrix composites reinforced by metal on the edges for improved resistance to foreign object damage and wear[1].

   Performance improvement and weight reduction have also been achieved by employing advanced design techniques. For many years, fan blades for aero-engines with high bypass ratios have been made from solid titanium alloys, which were designed with dampening snubbers at the mid span for vibration control. However, snubbers reduced aerodynamic efficiency, resulting in increased fuel consumption. Advanced designs have eliminated the snubber for greater aerodynamic efficiency and increased the blade chord length for greater mechanical stability. These design concepts reduced the number of blades by about one third and reduced the weight by employing a hollow construction. Typically, the hollow design has a low density core made of a honeycomb or corrugated material[2]. For example, honeycomb I-beam structures[3] and lightweight sound-absorbing honeycomb launch vehicle structures[4] have been proposed in the past. The core is an integral part of the structure which shares the centrifugal load, thus the panel-to-panel and core-to-panel joints must withstand foreign object impact and high cycle fatigue.

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Despite their proven history, current fan blade materials and designs suffer from high manufacturing costs, which add to the overall cost of the aircraft and rotorcraft. This high engine cost is especially limiting in terms of reduced aircraft sales in developing economies with limited financial resources. Recent advances in foam theory and manufacturing techniques have created a renewed interest in the application of metallic foams in the fabrication of engineering components [5-10]. Foams, which fall under the general category of cellular materials, provide several advantages to the aerospace designer due to their diverse multifunctional characteristics and uses. In particular, metallic foams possess low density, energy absorption capabilities and vibration dampening, which make them especially attractive for designing fan blades out of cheap and common materials, such as high strength and high toughness precipitation hardened (PH) stainless steels.

The present research was conceived under NASA's ULTRASAFE Project as a way of demonstrating the feasibility of designing and fabricating fan and propeller blades using a well known aerospace grade stainless steel, 17-4 PH. The 17-4 PH stainless steel was chosen due to its attractive mechanical properties and its relative ease of making foam cores from metallic powder. The overall goal of the research is to demonstrate that fan and propeller blades fabricated from a commonly available stainless steel using a combination of foam technology and innovative design concepts are comparable to currently-used titanium alloy or polymer matrix composite fans in their safety and performance criteria while being significantly cheaper. The proposed architecture for the fan or propeller blade system is a light weight sandwich construction made up of thin solid face sheets and a metallic foam core (see Fig. 1). The stiffness of the sandwich structure is increased by separating the two face sheets by a foam core. The resulting high-stiffness structure is lighter than that constructed only out of the solid metal material. Since the face sheets carry the applied in-plane and bending loads[5], the sandwich architecture is a viable engineering concept. However, the metal foam core must resist transverse shear loads while remaining integral with the face sheets. Other challenges relating to the fabrication and testing of these foam panels remain but these issues are outside the scope of the present paper. Instead, this paper presents vibration and stress analyses of the 17-4 PH sandwich fan blade with 17-4 PH metal foam core for different combinations of skin thickness and core volume.
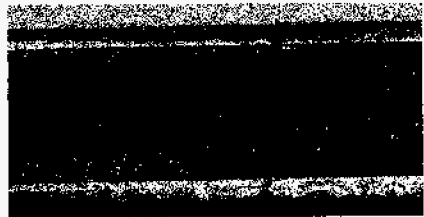

Figure 1: Side view of a 17-4 PH stainless steel foam core enclosed within two 17-4 PH face sheets.

II. Flexural Rigidity Design Criterion

Table 1 compares the material properties of wrought solid 17-4 PH stainless steel with a H-1025 heat treatment, with typical wrought Ti-6Al-4V blade material from Ref. 11. The elastic modulus of 17-4 PH is about 72% higher than titanium but with an added penalty of an increased density by about the same percentage. The other properties are in the same ranges. Hence, a 17-4 PH sandwich blade using a light weight metal foam core between two solid stiff face sheets is a viable design concept to reduce the blade weight without compromising the effective bending stiffness. The flexural rigidity of various rectangular sandwich foam constructions were determined as compared to a solid Ti-6Al-4V plate with identical dimensions. Figure 2 shows the ratio of the equivalent flexural rigidity $(EI)_{eq}$, defined as the ratio of (EI) for the 17-4 PH sandwiched construction to (EI) for a solid Ti-6Al-4V material, where E is the young modulus and I is the equivalent moment of inertia of the rectangular cross section. The equivalent (EI) ratio is plotted as a function of face sheet thickness, for an overall rectangular plate 160.0 mm long, 82.6 mm wide and 5.1 mm thick simulating an idealized experimental small scale fan blade geometry used at NASA Glenn Research Center for noise reduction studies. Since the total plate thickness is assumed constant and equal to 5.1 mm, the core thickness of the 17-4 PH sandwich plate decreases with increasing face sheet thickness by (5.1 mm − 2 t), where t is the face sheet thickness in mm. In the equivalent flexure rigidity

Table 1: Mechanical properties data comparison between 17-4PH and Ti-6Al-4V, from Ref. 11

		17-4PH H-1025	Ti-6Al-4V Grade 5
Modulus	GPa	196.50	113.76
density	g/cm³	7.64	4.43
Poison's Ratio		0.27	0.33
UTS	MPa	1276	1172
Yield	MPa	1172	1138
Fatigue-Limit	MPa	572	703

calculation, the foam elastic modulus E_c and shear modulus G_c were estimated using an analytical solution for open cell foam structures from Ref. 5:

$$E_c = E_s \left(\frac{\rho_c}{\rho_s} \right)^2 \qquad (1)$$

$$G_c = \frac{3}{8} E_s \left(\frac{\rho_c}{\rho_s} \right)^2 \qquad (2)$$

where E_s is the modulus of the solid material and (ρ_c/ρ_s) is the foam relative density (i.e. ratio of foam density to the solid material density). The foam relative density values considered here ranges from 6% to 24% relative density. The flexural rigidity of the sandwich 17-4 PH structure exceeds that of the solid Ti-6Al-4V plate for a face sheet thickness greater than 0.6 mm. The maximum flexural rigidity reaches as high as 1.72 times the Ti-6Al-4V for a face sheet thickness greater than 2.0 mm. Furthermore, the equivalent flexural rigidity ratio is independent of the foam relative density.
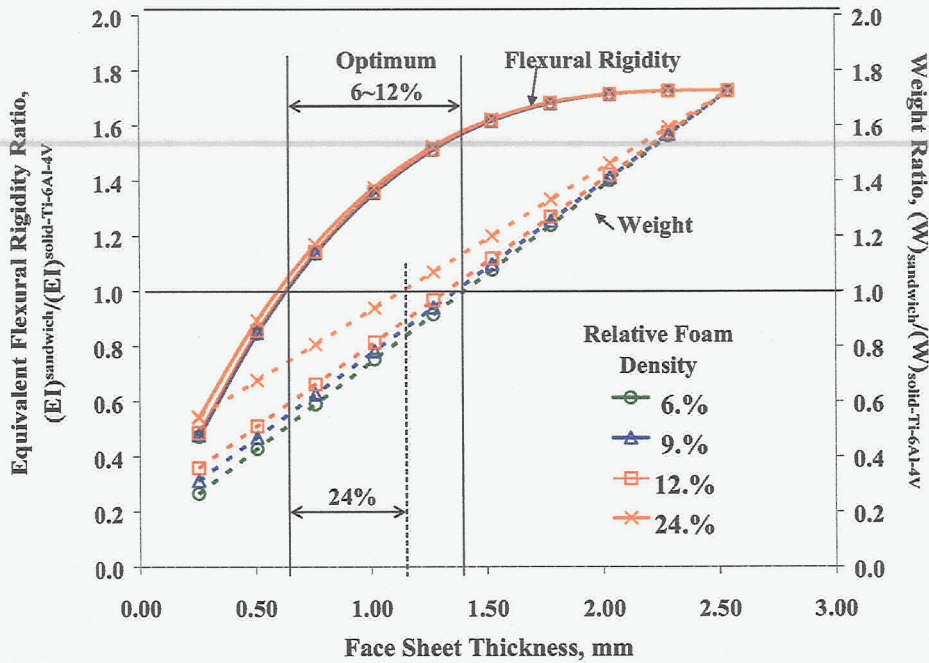
Figure 2: Variation of the equivalent flexural rigidity ratio and the weight ratio with face sheet thickness for a sandwiched 17-4 PH foam core plate of dimensions of 160.0 x 82.6 x 5.1 mm.

The weight ratio is also shown in Fig. 2, defined as the ratio of the sandwiched construction weight to the weight of a solid titanium plate. The curves clearly indicate that the 17-4 PH sandwich foam panel will be lighter than the solid titanium alloy for a face sheet thickness less than 1.4 mm for foam densities varying between 6 to 12%. However, the critical sheet thickness decreases to 1.2 mm for a relative density of 24%. Thus, the equivalent flexural rigidity ratio and the weight ratio plotted suggest that there exists an optimum 17-4 PH face sheet thickness range between 0.6 mm to 1.4 mm for an overall 5 mm plate thickness for a relative foam density of less than 12%. For the higher foam density (24%) the optimum range is between 0.6 mm and 1.2 mm. Hence, a sandwich blade construction is conceptually a feasible and competitive design for a fan blade if the flexural rigidity design is the only design criterion. But for a complete assessment of the feasibility for a sandwiched construction fan blade with metal foam core, other criteria need to be considered such as core failure, face wrinkling and face sheet debonding, to name a few[5]. Therefore, an experimental program was initiated to produce several 17-4 PH stainless steel sandwich foam panels.

III. Mechanical Properties Measurement of Sandwich Foam Panels

An initial batch of 17-4 PH sandwich panel with a metal foam core was procured from PORVAIR Inc.[12], Hendersonville, NC for microstructural and mechanical property determination. The foam cores were produced by a proprietary powder metallurgy technique and the face sheets were brazed to the foam with a BNi-6 type brazing alloy. The foam cores had a relative foam density of 6.0% with about 3 pores per mm (80 pores per inch). It should be noted that these manufacturing conditions are a starting point and do not necessarily represent an optimized state. In fact, process optimization is a portion of this overall experimental study, which is not covered in this paper. Typically, the dimensions of these panels were 254 x 152 x 13 mm, where the foam core and each face sheet were about 9.6 mm and 1.6 mm thick, respectively. As shown in Fig. 2, this sheet face thickness was chosen to give the sandwich foam panels the maximum

Figure 3: Optical micrograph showing the porosity in the17-4 PH foam ligaments. a) general foam microstructure, b) close up view of the foam ligaments.

equivalent flexural rigidity. Some of the panels were examined by ultrasonic spectroscopy and ultrasonic c-scans in order to evaluate the quality of the bonding of the face sheets with the foam cores[13]. Initial non-destructive and microstructural observations revealed that the foam ligaments contained a large percent of porosity and the bonding along the length and width of the sandwich panels were not consistent showing regions with a poor bond as well as regions with a strong bond. Figure 3-a) reveals the general microstructure of the foam core and Fig. 3-b) shows a detailed microstructure of the ligaments. As seen in Fig. 3-b), the larger triangulated voids are presumed to result from the plastic pre-form used to create the foam, and the smaller pores are due to poor sintering of the metal powder.
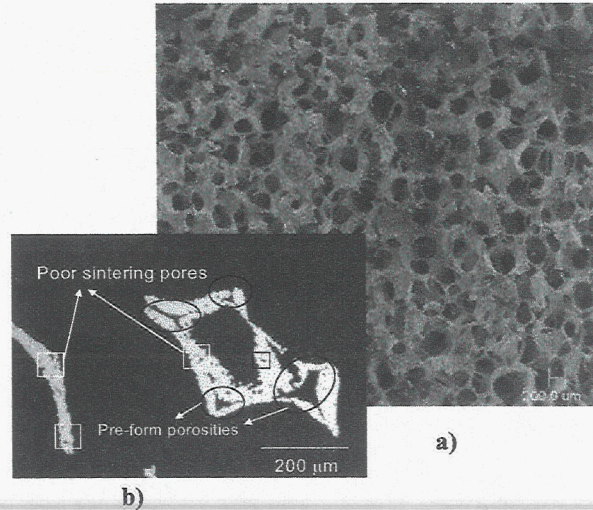
Specimens were also machined out of this initial batch of plates to determine the mechanical properties of the foam material. The experimental compressive stress-strain curves of two foam specimens are shown in Fig. 4. The specimens were periodically unloaded to get values for the unloading modulus as a function of strain. Also shown in the figure is a theoretical stress-strain curve for a defect-free foam structure. The stress-strain curves depict the three classical deformation regions characteristic of a cellular material[5]. The first stage, which is the linear elastic region, is followed by a second stage exhibiting a plateau region, known as the plastic collapse region, and finally by a third stage known as the
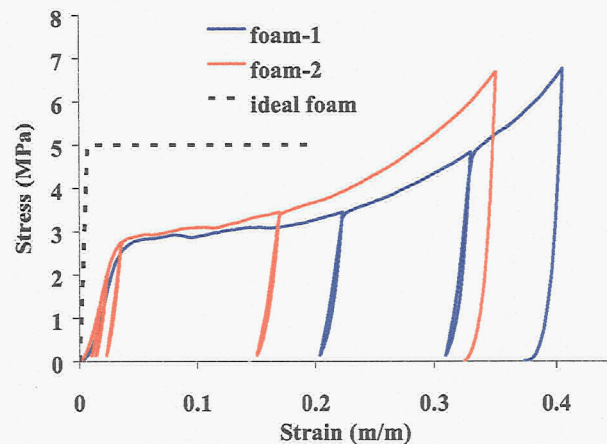

Figure 4: Compressive stress-strain curves at a displacement rate of 0.05 mm/sec for 17-4 PH foam with 6% relative foam density and 3 pores per mm.

densification region, where the foam core begins to densify with a corresponding increase in stress. The measured plastic collapse of the foam was of the order of 3 MPa, and the elastic modulus was 235 MPa, which are both lower than the theoretically calculated values of 5 MPa for a plastic collapse based on Ref. 5 and the 710 MPa ideal foam modulus from Eq. (1). Shear tests were performed on the sandwiched foam panels and gave an experimentally measured shear modulus in the range of 21 MPa, from Ref. 13, which is lower than the theoretically estimated value of 265 MPa using Eq. (2) by about an order of magnitude. This lower experimental shear modulus was attributed to poor interfacial bonding between the face sheets and the foam core. The average measured shear strength for the well-bonded specimens, where failure occurred in the core, was approximately 2.2 MPa, which is slightly lower than the measured compressive plastic collapse value of 3 MPa and about 50% lower than the theoretically estimated compressive plastic collapse value of 5 MPa. The poor comparison between the experimentally measured mechanical properties and the theoretical prediction can be attributed to the relatively large number of porosity observed in the cell ligaments, as seen in Fig. 3-b). In order to improve the mechanical performance of the foam, a process optimization study has been initiated with PORVAIR, Inc. to minimize the porosity in the cell ligaments and to improve the bond strength of the face sheets to the foam core. The analysis discussed in the next sections assume ideal mechanical properties for the foam core material with the anticipation that the process optimization study will minimize the disparity between the theoretical and experimental data.

IV.   Vibration Analysis

Part of the process of any blade design is to determine the natural frequencies of a particular blade shape. The designer requires a blade with sufficient stiffness so that its natural frequencies are high enough to avoid resonance within the engine operating range. Operating a blade at or near the resonant frequencies leads to high cycle fatigue which ultimately limits blade life. Vibration analyses on sandwich foam structures were conducted first on a simple rectangular geometry in order to gain some insights on the variation of the natural frequencies as a function of face sheet and core thickness. Next the vibrational characteristics of a complex shape fan blade with forward swept profiles shown in

Figure 5: a) NASA experimental fan blade and b) Idealized fan blade

Fig. 5-a), used at NASA Glenn Research Center in noise reduction studies[14], were determined assuming three different 17-4 PH constructions: a solid, a hollow, and a sandwiched structure. These results were then compared with a similar analysis performed on a Ti-6Al-4V solid blade of identical geometry and dimensions. The finite element method was used to determine the natural frequencies using the proven Lanczos method[15] to extract the first five natural frequencies and modal displacements.
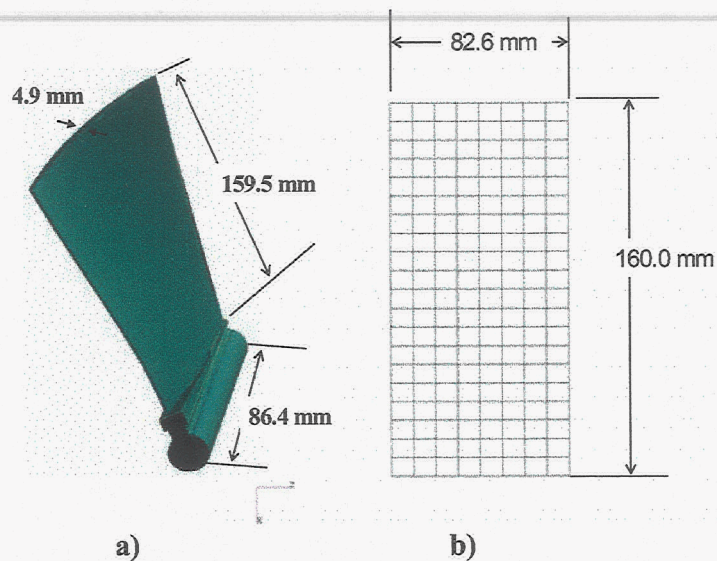
The modal shapes of the first five natural frequencies for the idealized blade configuration shown in Fig. 5-b (i.e., a flat plate) are shown in Fig. 6. The first and the third modes are the bending modes along the weak axis. The second and fifth modes are for twisting, and the fourth mode is for bending along the stiff axis. The variation of the natural frequencies with foam core thickness is shown in Fig. 7 for the 160.0 mm x 82.6 mm rectangular plate with a variable foam core thickness and a 1.4 mm constant face sheet thickness. The relative density of the 17-4 PH foam core was assumed to be 6.0%.
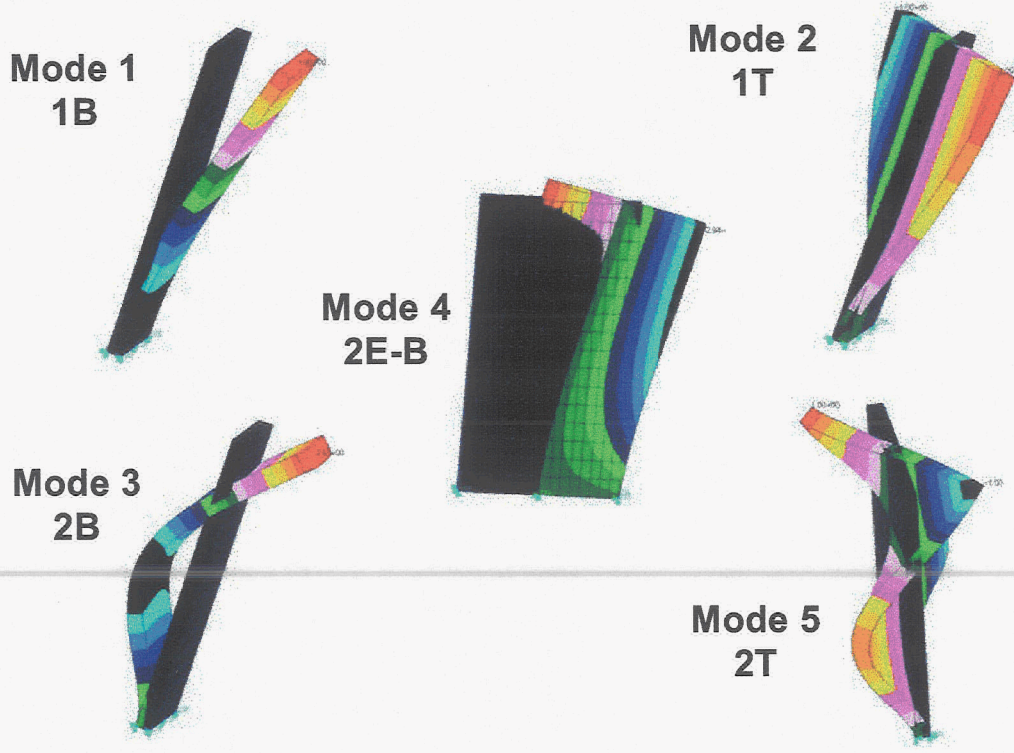
Figure 6: First five vibrational modal displacements for an idealized rectangular blade.
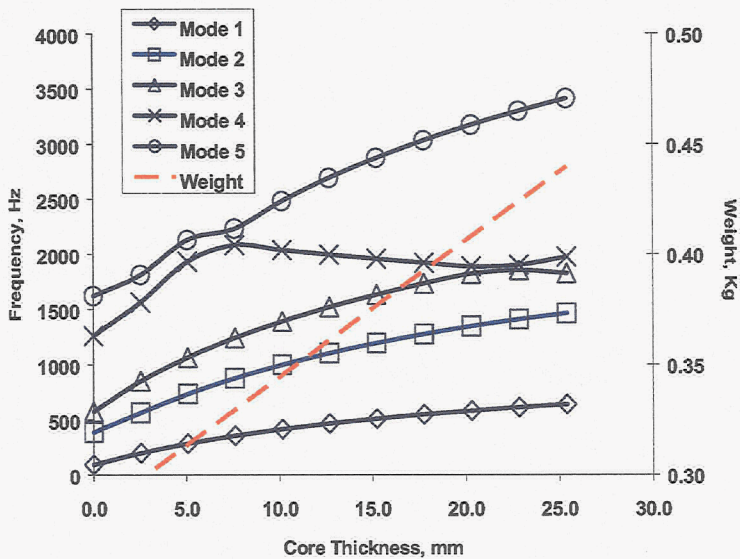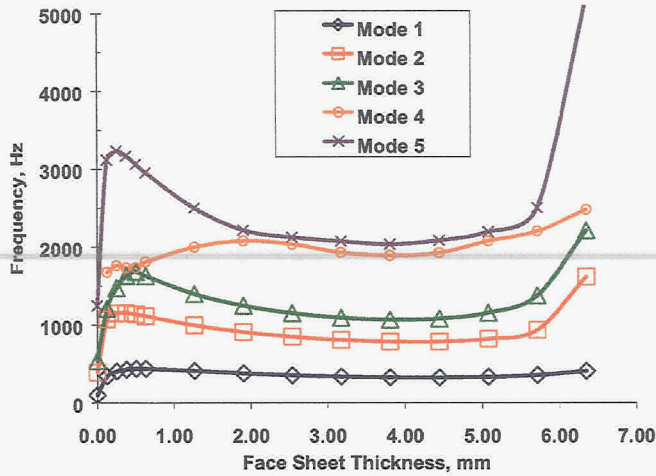

Figure 7: Variation of the first five natural frequencies as a function of the core thickness for the 160.0 mm x 82.6 mm plate having a constant 1.4 mm face sheet thickness.

As seen in Fig. 7, the first three modes corresponding to the first bending, first torsion and second bending modes, respectively, increase steadily with increasing the core thickness. The first mode increases from approximately 90 to 637 Hz by increasing the core thickness from 0 (i.e. solid plate) to 25.4 mm, which corresponds to a 600% increase in the frequency with a corresponding increase in weight of about 56%. While the second mode increased from 385 Hz to 1463Hz, the third mode increases from 573 Hz to 1824 Hz. The fourth mode, corresponding to the stiff bending mode, increases initially to reach a peak frequency of 2083 Hz at a core thickness of 7.6 mm and then declines steadily to 1900 Hz thereafter. Hence, the designer can easily increase the natural frequencies by just

increasing the light weight core thickness. Although this is beneficial for both ease of manufacturing the sandwich foam panels as well as for increasing the resonant vibration frequencies, the increase in overall thickness is likely to reduce the aerodynamic efficiency.

Figure 8 shows the variation of the natural frequencies with face sheet thickness keeping the overall plate thickness constant at 12.7 mm, which effectively decreases the ratio of the foam core thickness with increasing the face thickness. For the first bending mode (Mode 1), the natural frequency is almost constant. A similar trend is exhibited by the curve for the fourth mode. The second and third modal frequencies actually decreased sharply by introducing a foam core between the two face sheets, then starts increasing slightly with a reduction in the face sheet thickness until they drop again when only the foam material is used. A similar trend is seen at a larger scale for the fifth mode. The observed trend can be explained as a competition between the effective stiffness of the system and the effective mass. A reduction in the face sheet thickness that reduces the mass of the plate is more pronounced than a reduction in its stiffness, since the latter is primarily proportional to E.

Figure 8: Variation of the natural frequency as a function of the face sheet thicknesses for the 160.0 x 82.6 x 12.7 mm having an overall constant plate thickness.

Next, the first five natural frequencies of the fan blade profile shown in Fig. 5-a) were calculated using the finite element method for different blade constructions and materials. For all cases, the outer blade geometry was always the same and only the core is solid, hollow or filled with the metal foam. First a solid blade construction was assumed in order to compare the natural frequencies for a Ti-6Al-4V blade with that for a 17-4 PH solid blade. Then, two hollow blade 17-4 PH constructions were assumed with two different wall thicknesses of 0.73 mm and 1.81 mm. Finally, the two hollow 17-4 PH blades were filled with the 17-4 PH foam having ideal mechanical properties and a relative density of 6%.

Table 2 summarizes the natural frequencies and blade weights for various blade constructions. For the solid blades, the natural frequencies of a 17-4 PH solid blade are almost identical to that for a Ti-6Al-4V solid blade while its weight is 72% higher. These similarities in the frequencies are not surprising since the improvement in the stiffness of the 17-4 PH material to increase the frequency is eliminated by the increase in density given that the frequency is proportional to the ratio of modulus to density. For the hollow blade constructions, a reduction in the

Table 2: Natural frequencies for an experimental NASA fan blade and corresponding weight for various designs.

	Solid Blade, Hz		Hollow Blade (17-4 PH), Hz		Metal Foam Core (17-4 PH), Hz	
	Ti-6Al-4V	17-4 PH	0.73 mm skin	1.81 mm skin	0.73 mm skin	1.81 mm skin
Mode 1	335	332	210	297	332	317
Mode 2	1073	1080	420	756	990	985
Mode 3	1359	1363	821	1132	1239	1226
Mode 4	2390	2404	852	1528	2036	2056
Mode 5	2824	2840	1037	2111	2496	2569
Weight (kg)	0.294	0.507	0.137	0.311	0.159	0.323

frequencies is obvious especially for the higher frequencies. For the first mode the reduction in frequency of the hollow construction to a solid construction is about 37% and 11% for the 0.73 mm thin-walled blade and 1.81 mm thicker-walled blade, respectively. For the torsional frequencies the reductions in frequencies are 63% and 61% for Mode 2 and 5, comparing the thin walled ($t = 0.73$ mm) blade, to the solid blade. For the thicker walled blade, the reductions in the torsional frequencies are reduced by about 30% and 25%. When the hollow blades are filled with 17-4 PH metal foam, the reductions in the frequencies are insignificant compared to those of the solid blade with the largest deviation being 15% for Mode 4. The advantage of the foam core is observed by comparing the natural frequencies of Modes 2, 4 and 5 for the thin hollow walls as compared to the thin wall with the foam core. The observed improvement is greater than 130% for these modes with a minimal increase in weight. The improvement is less pronounced, although still significant, for the thicker walled blade reaching a 30% increase for Modes 2, 4 and 5. The smaller improvement is mainly due to the rigidity of the thicker face sheets in the hollow blade.

In summary, a solid fan blade design made out of 17-4 PH while having comparable vibrational characteristics to a solid Ti-6Al-4V blade suffers from a high weight penalty. At the other extreme, hollow fan blades made out of 17-4 PH, while leading to a considerable weight reduction comparable or lower than solid titanium alloy fan blades, possess low vibrational frequency modes. In contrast, 17-4 PH sandwiched fan blades with a foam metal core result in a much better vibrational performance over the hollow fan blade designs with weights comparable to or lower than a solid Ti-6-4 fan blade (Table 2). In this case, the decrease in the natural frequencies for the sandwiched foam designs over that of the solid Ti-6Al-4V fan blade is approximately 15% for Mode 4, but is typically less than 10% for all the other modes for the range of skin thickness shown in Table 2. This decrease in the natural frequency is likely to be acceptable given the potential for tremendous cost savings in using 17-4 PH instead of Ti-6Al-4V. Additionally, the weight savings of a sandwiched blade compared to a solid titanium alloy blade is about 46% when the skin thickness is 0.7 mm.

V.   Stress Analysis

Finite element analyses were performed in order to compare the displacements and stresses for various blade constructions of the NASA Glenn experimental fan blade geometry[14] (see Fig. 5-a) assuming a pure elastic condition. The constructions considered in this section are a solid Ti-6Al-4V and 17-4 PH blade, and 1.81 mm 17-4 PH thick-walled hollow and foam filled sandwich constructions, all having the same outer blade dimensions. Three loading conditions were assumed for all four blades. The first loading is pure rotational at 3,000 rpm, with all the blade root nodes assumed fixed in all three directions. The second loading is a combination of the rotational loading and an applied pressure along the blade profile. The third loading condition is a combination of the second loading condition with an additional distributed load at the blade tip simulating a bird impact. The impact area assumed is a 22 x 15 mm rectangular area (highlighted in Fig. 9) at the tip of the leading edge with a 5 kN force for a 0.1 msec duration.
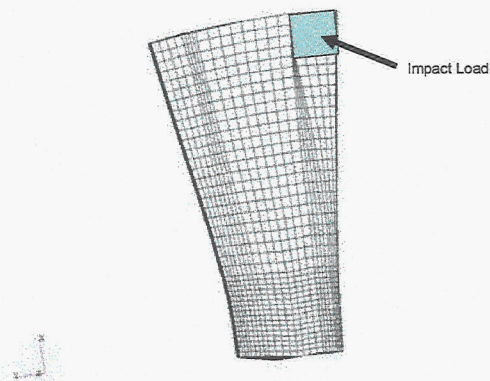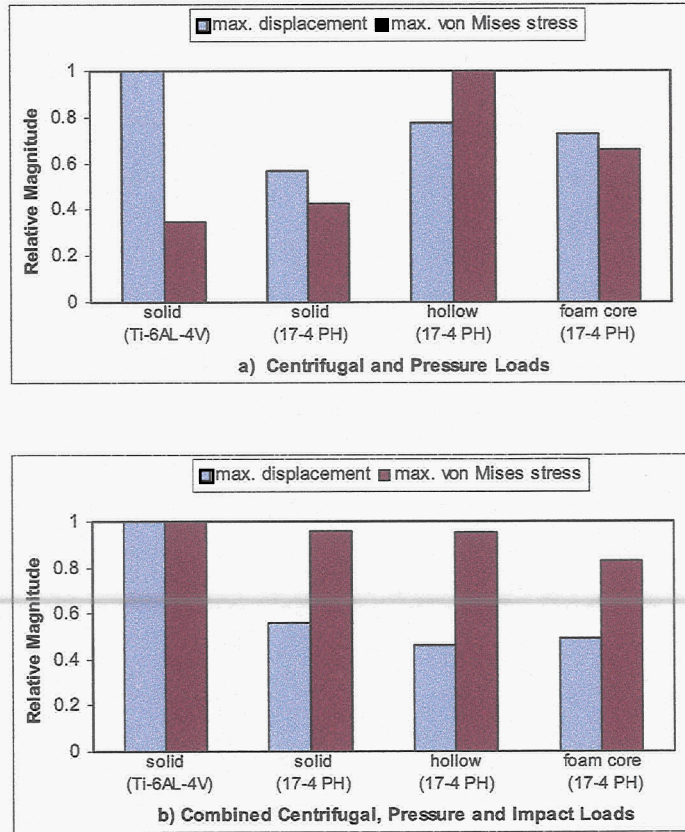

Figure 9: NASA Glenn experimental fan blade mesh highlighting the size and location of the impact load.

Figure 10 shows a relative comparison of the maximum stress and maximum displacements for the four blade designs. Table 3 lists the maximum displacement magnitudes and von Mises equivalent stresses for the four blade constructions and the three loading conditions. The maximum von Mises stresses in Table 3 occur in the outer fibers of the blade and correspond in the foam construction to the stresses in the face sheets. The maximum displacements due to a pure rotational loading are almost identical, with displacements in the hollow 1.81 mm 17-4 PH blade being slightly larger. Under combined pressure and rotational loadings, the maximum displacement magnitudes were largest with the solid Ti-6Al-4V followed by the hollow 17-4 PH blade, the sandwiched foam blade and finally with the solid 17-4 PH. These results can be easily explained by the effects of the variation in

flexure rigidity of the various blade constructions. The solid Ti-6Al-4V blade with its low modulus had the highest displacement magnitude, the solid 17-4 PH had the smallest deflection due to its high modulus, and the hollow and sandwiched foam blades fell in the middle. The maximum stresses in the hollow and sandwiched foam blades are higher for both pure rotation as well as a combination of rotation and pressure compared to the stresses in the solid Ti-6Al-4V and 17-4 PH. The solid 17-4 PH was expected to result in the highest stress value due to its weight, but the actual calculated values were lower than for the hollow and sandwich foam constructions. It is postulated that due to the blade curvature and geometry, the loading observed by the blade is not uniaxial tension, but has some bending and twisting components, that causes the hollow and sandwiched blade constructions to result in higher stresses. For the combined rotation and pressure loadings, the maximum stress is observed in the hollow blade, followed by the sandwiched foam, then the solid 17-4 PH and finally the solid Ti-6Al-4V blade. A distressing observation is that the maximum von Mises equivalent stress in the sandwiched foam blade, although lower than the hollow blade, is almost double the solid Ti-6Al-4V blade stress value.

Figure 10: Comparison of relative magnitudes of maximum displacement and stress for various blade designs.

Table 3: Maximum displacement magnitudes and von Mises equivalent stresses for the NASA Glenn experimental fan blade geometry under various load combinations.

Blade Construction Type	Blade Material	Face Sheet Thickness	Pure Rotation		Combined Rotation & Pressure		Combined Rotation, Pressure and Impact	
			Max. Displacement Magnitude	Max. von Mises Stress	Max. Displacement Magnitude	Max. von Mises Stress	Max. Displacement Magnitude	Max. von Mises Stress
		mm	mm	MPa	mm	MPa	mm	MPa
Solid	Ti-6-4	N/A	0.094	27	1.000	64	3.373	422
Solid	17-4 PH	N/A	0.094	43	0.566	79	1.891	404
Hollow	17-4 PH	1.81	0.108	86	0.774	185	1.546	401
Sandwich with Foam Core	17-4 PH	1.81	0.099	73	0.727	122	1.646	349

With the addition of the impact loading, the maximum displacement occurred for the solid Ti-6Al-4V blade, while that for the foam blade was the second lowest. Furthermore, the von Mises stress under a combined rotation, pressure and impact loading was also the largest for the Ti-6Al-4V solid blade and the least for the 17-4 PH

sandwiched foam blade. The stresses for the hollow and solid 17-4 PH under the combined loading with impact were intermediate.

Although the maximum stresses for the sandwiched foam construction occurred in the face sheets for the various load combinations, the lower stresses in the foam core can not be ignored. A careful inspection of the resulting stresses in the foam core revealed that relatively high stresses, that are almost an order of magnitude higher than the experimentally measured plastic collapse stress value, are observed mainly along the apex of the foam core as it meets the thin face sheets at the leading and trailing edges. A smooth radius in these areas is expected to reduce these observed stress risers.

VI. Conclusion

A study was conducted to provide an assessment on the 17-4PH stainless steel foam sandwiched fan blade design over currently used solid titanium alloy blades. An optimum design criterion for the 17-4 PH stainless steel core foam sandwich construction was determined for better flexural rigidity over a solid titanium blade while substantially reducing weight. The vibration analysis results showed that significant weight/cost savings can be achieved with acceptable natural frequencies over a solid titanium alloy blade. The stress analysis revealed that the stresses developed along the face sheets for a sandwich construction are higher than the solid blade under stress conditions resulting from rotation and applied pressure. However, under a combined impact, rotation, and pressure loading condition, the 17-4 PH blade construction with a foam core resulted in the lowest von Mises equivalent stress in the outer fiber (face sheets) compared to all the other blade constructions. The maximum displacement of the foam core blade was also lower than the solid Ti-6Al-4V blade, clearly demonstrating the advantages of the metal foam construction. In conclusion, a sandwich blade construction of a fan blade with metal foam core built from a common aerospace grade 17-4PH stainless steel is a competitive design to achieve the similar levels of performance criteria while being substantially cheaper for advanced aircraft fan and rotorcraft propeller blades development.

References

[1]"GE 90 Fan Blade," URL: http://www.geae.com/ourcommitment/innovation/ge90fanblade.html.

[2]Stewart, M., "Advanced materials mean advanced engines," Materials World, vol. 4, 1996, pp. 446-449.

[3]Mullen, S. J., "I-Beam Honeycomb Material", U. S. Patent No. 4,632,862, 1986.

[4]Newton, J. F., Martin, T. D., Willard, N., Carbery, D. J., and Ikegami, R., "Lightweight Honeycomb Panel Structure," U. S. Patent No. 5,445,861, 1995.

[5]Gibson, L. J. and Ashby, M. F., *Cellular Solids: Structure and Properties*, Cambridge University Press, Cambridge, U.K., 1997.

[6]Harte A.M., Fleck N.A. and Ashby M.F., "Sandwich Panel Design Using Aluminum Alloy Foam," *Advanced Engineering Materials*, Vol. 2, No 4, 2000, pp. 219-222.

[7]Ashby M.F., Evans A.G., Fleck N.A., Gibson L.J., Hutchinson J.W. and Wadley H.N.G., *Metal Foams: A Design Guide*, Butterworth-Heinemann, Boston, 2000.

[8]Banhart J., "Manufacture, Characterisation and Application of Cellular Metals and Metal Foams," *Progress in Materials Science*, Vol. 46, 2001, pp. 559–632.

[9]McCormick, T. M., Miller, R., Kesler, O. and Gibson, L. J., "Failure of Sandwich Beams with Metallic Foam Cores," *Intern. J. Solids and Structures*, Vol. 38, No. 28-29, 2001, pp. 4901-4920.

[10]Wallach J. C. and Gibson, L. J., "Mechanical Behavior of a Three-Dimensional Truss Material," *Intern. J. Solids and Structures*, Vol. 38, No. 40-41, 2001, pp. 7181-7196.

[11]*Structural Alloys Handbook, 1996 edition*, John M. (Tim) Holt, Technical Ed; C. Y. Ho, Ed., CINDAS/Purdue University, West Lafayette, IN, 1996.

[12]PORVAIR, Fuel Cell Technology Division, Hendersonville, NC, USA.

[13]Gosgriff, L.M. et. al. "Ultrasonic Spectroscopy of Stainless Steel Sandwich Panels," Presented at the SAMPE 2003 "Materials and Processing: Enabling Flight … Our Legacy and Future" Conference, September 28-October 2, 2003, Dayton, Ohio.

[14]NASA-Glenn Research Center: Quiet Aircraft Technology (QAT) Efficient Low Noise Fan (ELNF) Critical Design Review (CDR), July 2003.

[15]Hughes, T. J.R., The Finite Element Method: Linear Static and Dynamic Finite element Analysis, Prentice-Hall, Inc., 1987, pp. 582-592.