THERMO-MECHANICAL ANALYSIS OF THE HIBACHI FOIL FOR THE ELECTRA LASER SYSTEM

Aaron Aoyama[1], James Blanchard[2], John Sethian[3], Nasr Ghoniem[1], and Shahram Sharafat[1]

[1]*University of California Los Angeles: 420 Westwood Plaza, Los Angeles, CA, 90095, takeo556@gmail.com*
[2]*University of Wisconsin: 1513 University Avenue, Madison, WI, 53706*
[3]*Naval Research Laboratory: 4555 Overlook Avenue, Washington, DC, 20375*

In support of the High Average Power Laser (HAPL) project the Electra Laser, a KrF Gas Laser system is being developed at NRL. The laser uses high voltage (500 – 800 keV), high current (100 – 500 kA), short pulse (100 – 600 ns) electron beams to pump the 0.14 MPa (20 psi) pressurized KrF gas cell, which is separated from the vacuum region by a 25 µm-thick stainless steel foil, the Hibachi Foil. The foil is made of SUS304, operates between 180 ℃ and 450 ℃, and has typical dimensions of about 0.3 m × 1.0 m. The laser pulses at up to 5 Hz, and the foil is subjected to repetitive thermal and mechanical stresses. In typical experiments, the foil lasts 1000 – 20,000 shots before suffering a catastrophic failure. In an attempt to improve foil performance a variety of design modifications are being considered along with changes in foil material. Earlier Hibachi foil designs used flat foils resting on 0.3 m long square water-cooled supporting ribs (1 cm wide). There is a 3.4 cm gap between ribs. . Advanced Hibachi foil concepts are under development using a scalloped foil design. In this paper we report on the comparative thermo-mechanical analysis between flat and scalloped foil geometries. It is demonstrated that the scalloped design reduces stresses to within yield limits of the stainless steel material.

I. INTRODUCTION

The Electra Laser is a KrF Gas Laser system being developed at the Naval Research Laboratory (NRL) for use in the High Average Power Laser (HAPL)[1] project. In the laser system, a large area electron beam is generated within a vacuum region. The electron beam is passed through a 25 µm-thick stainless steel foil into the KrF gas cell pressurized to 0.14 MPa (20 psi). As the laser pulses at frequencies up to 5 Hz, the foil undergoes repetitive thermal and mechanical stresses. Under these conditions, the energy deposited by the electron beam causes the foil to reach an average operating temperature of 250ºC. The KrF gas also heats up enough to cause a pressure increase of 0.043 MPa (6.35 psi) while the heated volume is kept sealed. Because the real setup allows the gas to flow openly, only a fraction of this pressure increase would actually be felt by the foil. Since the real pressure felt by the foil is unknown, the analysis presented in this paper assumes the foil feels the full 0.043 MPa cyclic pressure load. Experimental results show the foil lasting 1,000-16,000 laser shots before catastrophic failure. The desired operating lifetime is several orders of magnitude longer.[2]

In an effort to improve foil performance, a variety of design modifications and material changes are being considered. While early Hibachi foil designs consisted of flat foils supported by square-shaped ribs, more advanced concepts using different supporting rib and foil shapes are being developed.

In this paper we report on the thermo-mechanical analyses of different foil geometries, and perform a comparative study in order to work towards an optimal design.

I.A. Background

Fig. 1 shows the overall electron beam and foil configuration. The electron beam is generated in the vacuum region over an area 30 cm high and 100 cm wide. The foil is supported by an array of ribs as shown. Because the ribs look like a barbeque grill, the support structure is known as the “hibachi” and the foil as the “hibachi foil.”

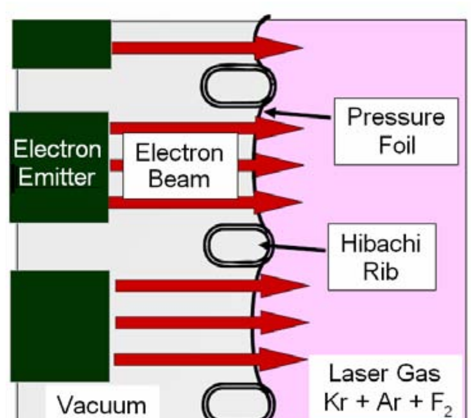

Fig. 1. Hibachi configuration.[2]

In order to maximize the electron transmission and foil heating, the hibachi foil thickness must be minimized. The foil must also resist the effects of:

- Atomic fluorine.
- Molecular fluorine.
- The hydrostatic pressure of the laser gas.
- The dynamic pressure and shock induced by sudden deposition of electron beam energy into the laser gas.
- The sudden heat pulse induced by a portion of the electron beam energy stopped in the foil.
- Ultra-violet light from the laser.
- X-rays from the electron beam (E<500 keV).
- Operation between 200 ºC and 450 ºC.

I.B. Global Performance Statistics

Tests were performed by NRL in order to help determine the failure modes and mechanisms of the foil. Between 5/13/2004 and 1/16/2008, 63 different foil samples were tested using a total of 447,963 laser shots. 33 foil failures occurred with 18 due to holes, and 15 due to blows. Heat induced wrinkles were found in 10 samples, and absent in 19. The remaining 4 had unknown wrinkle conditions. The maximum number of shots a sample withstood before failure was ~25,000 on a sample without heat wrinkles. Figs. 2 and 3 show the foil lifetime, and failure criteria respectively.

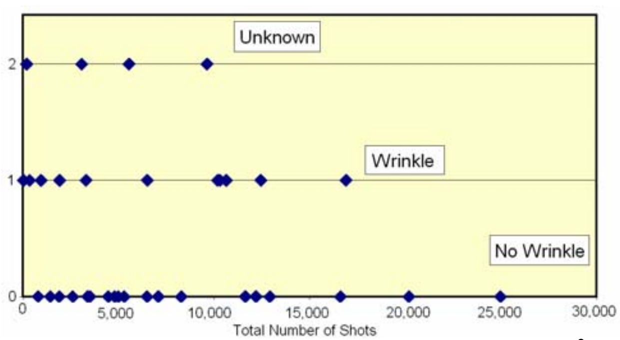

Fig. 2. Foil sample lifetime classified by heat wrinkle.[2]

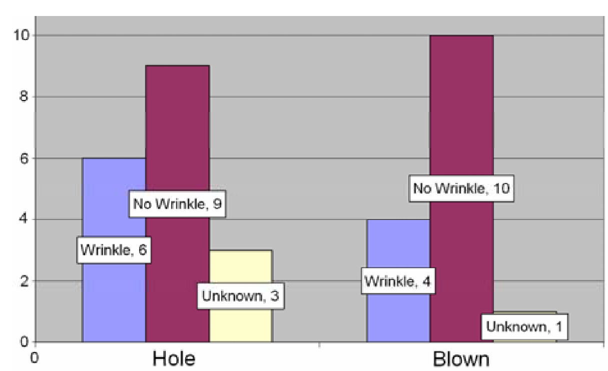

Fig. 3. Foil failure classified by failure mode and heat wrinkle condition.[2]

II. THERMO-STRUCTURAL ANALYIS

Finite element analysis (FEA) is performed on a variety of geometrical models and analysis types using ANSYS Multiphysics[3] and COSMOSWorks[4] in an effort to evaluate the merits of the different proposed designs. Four design variations were analyzed: flat foil with flat supporting ribs, foil with pre-loading curvature and flat support ribs, flat foil with curved support ribs, and a scalloped design with both foil and supports curved. Shell elements were used for elastic and plastic analysis.

II.A. Material Properties

For the finite element analysis, rate-independent, bilinear isotropic plasticity material models were used. Material properties were determined based on data provided by NRL. The Young's Modulus (E) and Tangent Modulus (E_{tan}) were estimated using the stress-strain data for SUS304 foil of 25-μm thickness. Fig. 4 shows stress strain data for new ("as received") and used ("exposed") SUS304 material at 21 ºC and 400 ºC.

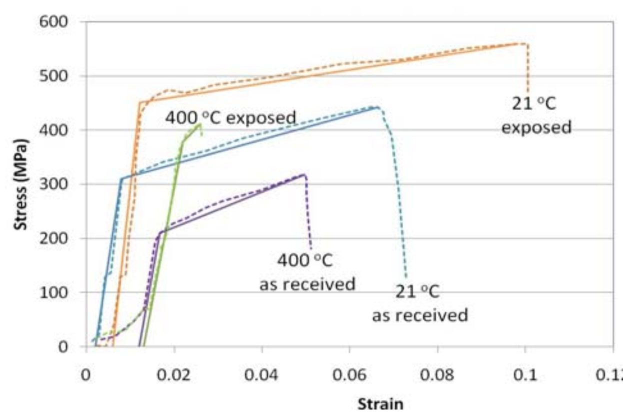

Fig. 4. SUS304 foil (25 μm thick) stress strain data (dotted lines: NRL data; solid lines: estimates[5])

Table I lists the estimated material properties for SUS304 foil. The yield stresses listed are provided directly by NRL[2].

TABLE I. Material Properties for SUS304 Foil

Test Temp. & Condition	E (GPa)	E_{tan} (GPa)	Yield Stress (MPa)
21 ºC as received	39.16	2.26	310
21 ºC exposed	36.82	1.27	434
400 ºC as received	12.59	3.29	209
400 ºC exposed	17.22	8.41	365

II.B. Flat Foil with Flat Support Ribs

The first finite element analysis deals with a flat foil with flat support ribs. The CAD model was generated using the dimensions of a single foil-rib pair as shown in Fig. 5. The foil is 30 cm long with a 3.4 cm gap between two 1 cm supporting ribs.

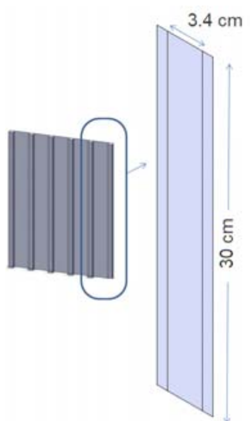

Fig. 5. Flat foil analysis CAD modeling.

All four outer edges are fixed, and pressure is applied normal to all faces. Under cyclic pressure loading between 0.14 and 0.183 MPa at 21 ºC, the model deforms elastically as shown in Fig. 6 with a maximum Von Mises stress of ~338 MPa, and ~3.9% strain along the rib edges.

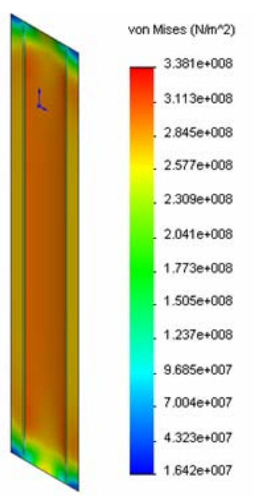

Fig. 6. Von Mises stress for flat foil at 0.183 MPa pressure and 21 ºC temperature.

When applying the same cyclic pressure loading after the foil is heated to 250 ºC, heat wrinkles develop parallel to the width of the foil. The resultant maximum Von Mises stress and strain are ~342 MPa and ~5%. This stress is close to the 392 MPa yield stress at 250 ºC for the exposed foil. The heat wrinkles can be seen in the isometric and front views of the Von Mises stress contour plots given in Fig. 7. These heat wrinkles are most prominent while the foil is held at elevated temperatures and pressures. However, due to residual stress effects, portions of the larger wrinkles still remain when the pressure and heat is removed, but only to a limited extent.

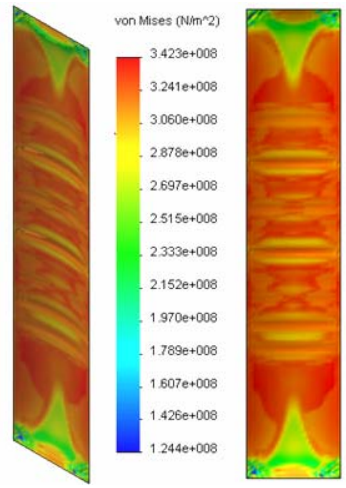

Fig. 7. Von Mises stress for flat foil at 0.183 MPa and 250 ºC (note formation of "heat wrinkles").

II.C. Curved Foil with Flat Support Ribs

To further examine the high stresses and strains along the lengthwise edges of the foil, a surface with curvature along the width was modeled as shown in Fig. 8.

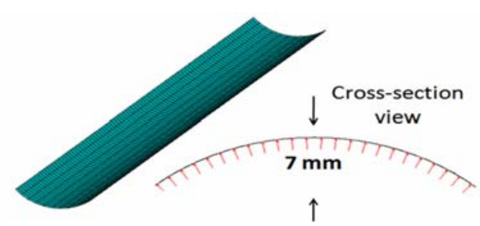

Fig. 8. Isometric and cross-sectional view of curved foil.

Under static 0.14 MPa pressure-only loading with all four edges fixes, the model deforms elastically. However, when temperature loading of 90 ^{o}C is added, a localized single wrinkle is caused by the thermal expansion as shown in Fig. 9.

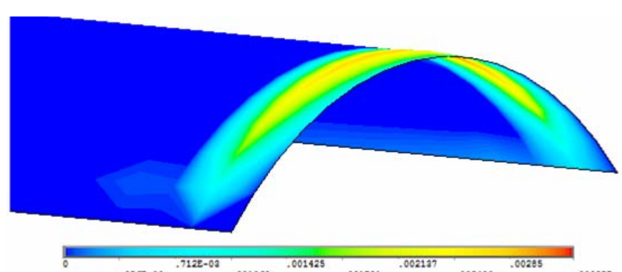

Fig. 9. Plastic strain contours of curved foil under thermal and pressure loading.

In order to determine if this localized heat wrinkle is caused by buckling, a series of buckling analyses were performed on the same model. A conservative eigenvalue buckling analysis was performed to determine an appropriate first buckling mode. From the displacement contours seen in Fig. 10, it appears that the largest deflections are near the ends where the foil buckles under temperature loading.

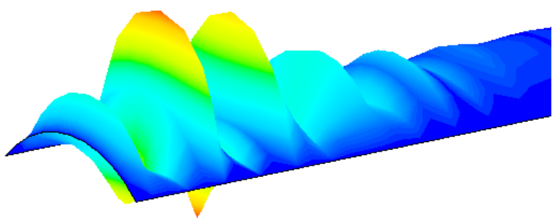

Fig. 10. Displacement magnitude contours of curved foil for eigenvalue buckling analysis.

A more conservative approach is to use a nonlinear buckling analysis. In this analysis case, the temperature load is increased until the solution begins to diverge. When divergence occurs, the ANSYS nonlinear stabilization option adds an artificial damper to maintain a stable state. The dampening coefficients are tracked and used to make corrections to the results. Fig. 11 shows how the thermal deformations are spread out more uniformly along the strip in the nonlinear analysis. The wrinkles remain after heat and pressure loads have been removed, which are observed both experimentally and in the analysis.

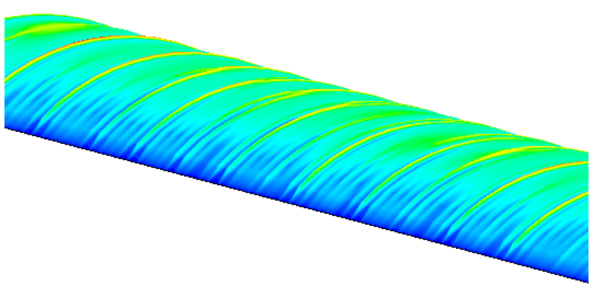

Fig. 11. Displacement magnitude contours of curved foil for nonlinear buckling analysis.

Due to the discrepancies between the different analyses performed on the curved foil model, it is too early to make any conclusions about which set of results is correct. Further work must be done on this analysis.

II.D. Flat Foil with Curved Support Ribs

In an attempt to reduce the plastic strain which occurs at the sharp corner at the edge of the support ribs, the third analysis investigates the possibility of a flat foil with curved support ribs. By avoiding sharp corners, this model attempts to alleviate the very large rotations of the foil about the long edges where it meets the support ribs. Fig. 12 shows the model used for this analysis.

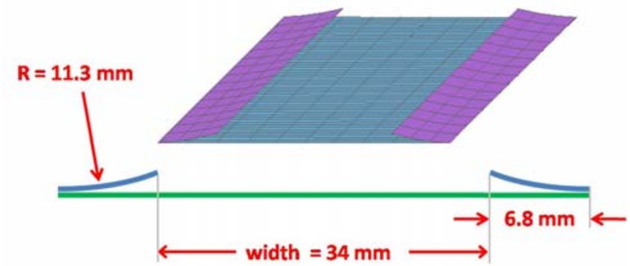

Fig. 12. Flat foil with curved support rib model.

Results show that the curved, rigid supports ease the transition to the foil's equilibrium state. The displacement plot in Fig. 13 shows how the foil deflects more gradually along the shape of the curved support ribs, avoiding the sharp deflection that occurs with flat supports. Additionally, flat foil is easier to work with than foil with a pre-loading curvature.

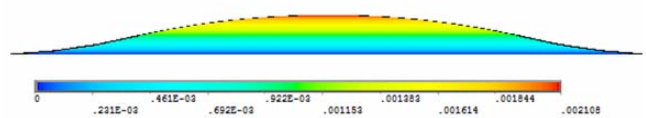

Fig. 13. Displacement magnitude contours of flat foil with curved support ribs.

The maximum plastic strain under loading (90 ^{o}C and 0.14 MPa) is ~1.6% for the flat foil with curved supports. An analysis of the original flat foil under the same loading conditions and similar mesh density results in a maximum plastic strain of ~3.9%. This shows that a significant decrease in plastic strain occurs when using curved supports instead of flat ones.

II.E. Scalloped Model with Curved Foil and Ribs

The final design examined is a scalloped model which features curved foil and curved supporting ribs.

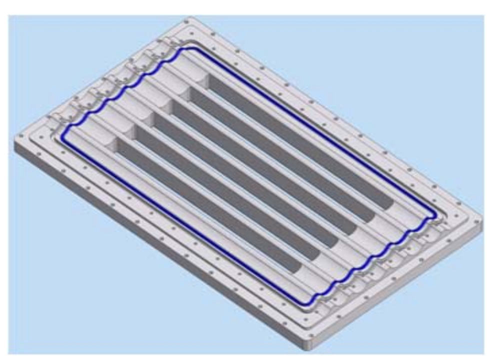

Fig. 14. Scalloped foil test section.[6]

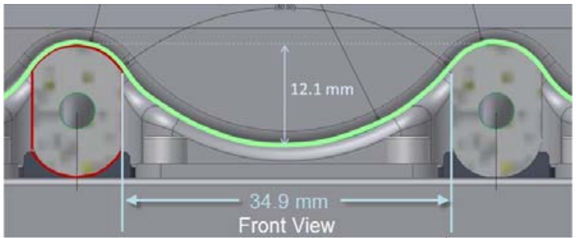

Fig. 15. Details of the scalloped foil model.[6]

This model attempts to further alleviate the rotation of the foil about the long edges. Figs. 14 and 15 show the scalloped foil model geometry that is under development. From these design details, a section of the curved foil and support ribs is modeled for finite element analysis as shown in Fig. 16.

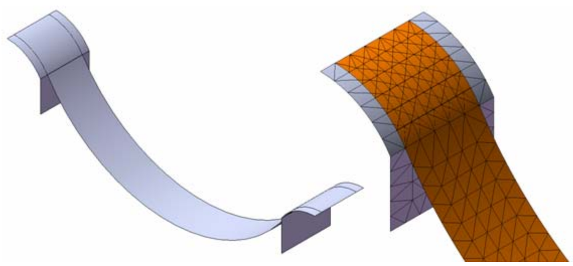
Fig. 16. FEA model and mesh for scalloped foil model.

This nonlinear pressure analysis assumes that the foil can slide along the curved ribs, necessitating a smaller section size in order to reduce the number of elements. The small section is modeled using symmetry boundary conditions. End effects of the foil geometry, which play an important role in the stress state, are absent and must be investigated in the future.

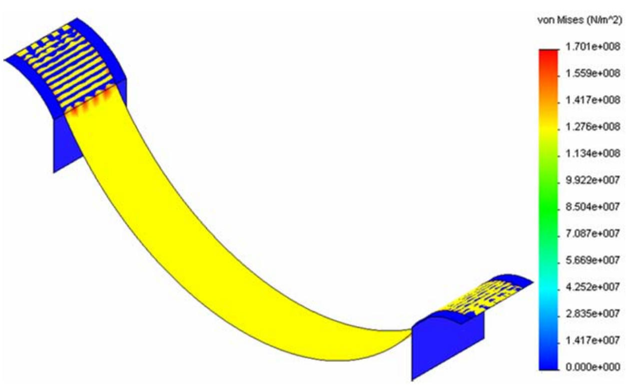

Fig. 17. Von Mises stress contours for scalloped model.

At a maximum pressure of 0.183 MPa with no temperature loading, results show small displacement magnitudes of ~30 μm, and plastic strain magnitudes of ~0.05%. Fig. 17 shows the maximum Von Mises stress to be ~170 MPa along the support edges. With this geometry and only cyclic pressure loading, the foil deforms elastically with strains and deformations which are magnitudes smaller than with the previous models. Additionally, the residual stress levels are a low 0.065 MPa as shown in Fig. 18, which is a factor of 330 less than the 21.44 MPa residual stresses found in the flat foil with flat supporting ribs model.

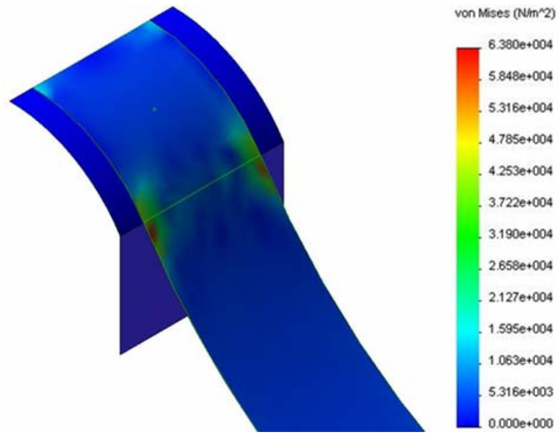

Fig. 18. Residual stress for unloaded scalloped model

III. SUMMARY AND CONCLUSIONS

This paper investigated four different hibachi foil designs. The flat foil with flat supports shows the expected heat induced wrinkle formation under pressure and thermal loading. Inconclusive results show that additional work needs to be done on the curved foil with flat support model. The pressure only analysis featuring flat foil with curved support ribs results in roughly 50% reduced strain when compared to the original flat foil/support model.

Finally, the pressure only analysis of the scalloped foil model shows a maximum displacement and strain ~100 times less than the original flat foil/support model, with roughly half the stress. The effects of thermal loading and end-geometry for the scalloped model still must be analyzed.

These initial analyses suggest that the performance lifetime of the hibachi foil structure may be improved by avoiding the sharp bends which occur at the ends of the original flat support ribs. Curved support ribs in particular appear to reduce the induced stresses and strains by alleviating the foil rotation at the support edges.

ACKNOWLEDGMENTS

This work is sponsored by the US Department of Energy, NNSA/DP and Naval Research Laboratories.

REFERENCES

1. JOHN D. SETHIAN, A. RENE RAFFRAY, JEFFERY LATKOWSKI, JAMES P.

BLANCHARD, LANCE SNEAD, TIMOTHY J. RENK, SHAHRAM SHARAFAT, An overview of the development of the first wall and other principal components of a laser fusion power plant, J. Nucl. Mater. 347: 3 (2005) 161-177

2. J. SETHIAN, F. HEGELER, J. GIULIANI and T. ALBERT, "The Hibachi Foil challenge from a materials/mechanical perspective," Technical Memo, Naval Research Laboratory, Washington, DC 20375, Jan. 11, 2008.
3. ANSYS® Academic Research product, http://www.ansys.com/corporate/contact.asp
4. SolidWorks/COSMOSworks Educational Products, http://www.solidworks.com/sw/engineering-education-software.htm
5. S. SHARAFAT, Mech. & Aerospace Engr. Dept., University of California Los Angeles, private communication, February 2008.
6. J. PARISH, Commonwealth Technology Incorporated, Alexandria, Virginia 22315, private communication, March 2008.