

RECTANGULAR PELLICLE BEAM SPLITTER DESIGN

by

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ABSTRACT

This project investigates a pellicle beam splitter of rectangular form. A pellicle is a thin optical membrane and a beam splitter separates one optical path into two. The beam splitter under investigation is used at Arnold Engineering Development Center (AEDC) for optical component testing and mission simulation in cryogenic vacuum chambers. The conventional beam splitter had undesirable optical performance at cryogenic temperatures. The goal of this project was to use analysis to guide development of pellicle beam splitter prototypes to minimize thermal distortion.

Engineers commonly use finite element analysis (FEA) to model structural performance. FEA models were created that represented various prototypes including dimensions, materials, and loads. The models had a rectangular pellicle with a fixed outer boundary. An inner ring of rectangular form was then pressed against the pellicle to create an optical aperture inside the inner ring. Then geometric parameters were varied one at a time to evaluate their effects. The parameters included inner ring corner radii, outer boundary corner radii, inner ring to outer boundary spacing, pellicle thickness, and inner ring displacement.

The FEA models indicated that the rectangular form of the pellicle beam splitter contributed to undesirable optical characteristics. Sharp corners of the inner ring create high stress concentration areas causing varying pellicle thickness and undesirable optical quality. Modeling showed that increasing inner ring corner radii and inner ring to outer boundary spacing are effective at increasing optical quality. Other parameters had little to negative affects on optical quality.

The rectangular shape of the pellicle beam splitter leads to a non-uniform stress distribution and therefore undesirable optical quality. Stress non-uniformity can be reduced but not eliminated by using large inner ring corner radii and large inner ring to outer boundary spacing. A prototype with large inner ring corner radii was made and tested. The prototype had much better optical performance with little corner affects as compared to previous prototypes.

INTRODUCTION / BACKGROUND

This project investigates a pellicle beam splitter of rectangular form. The pellicle beam splitter is created by mounting a pellicle, a thin optical membrane, to an outer rectangular frame. An inner tensioning ring is ground optically flat and pressed against the pellicle, defining an optically flat pellicle boundary. A prototype pellicle beam splitter can be seen in Figure 1.1.

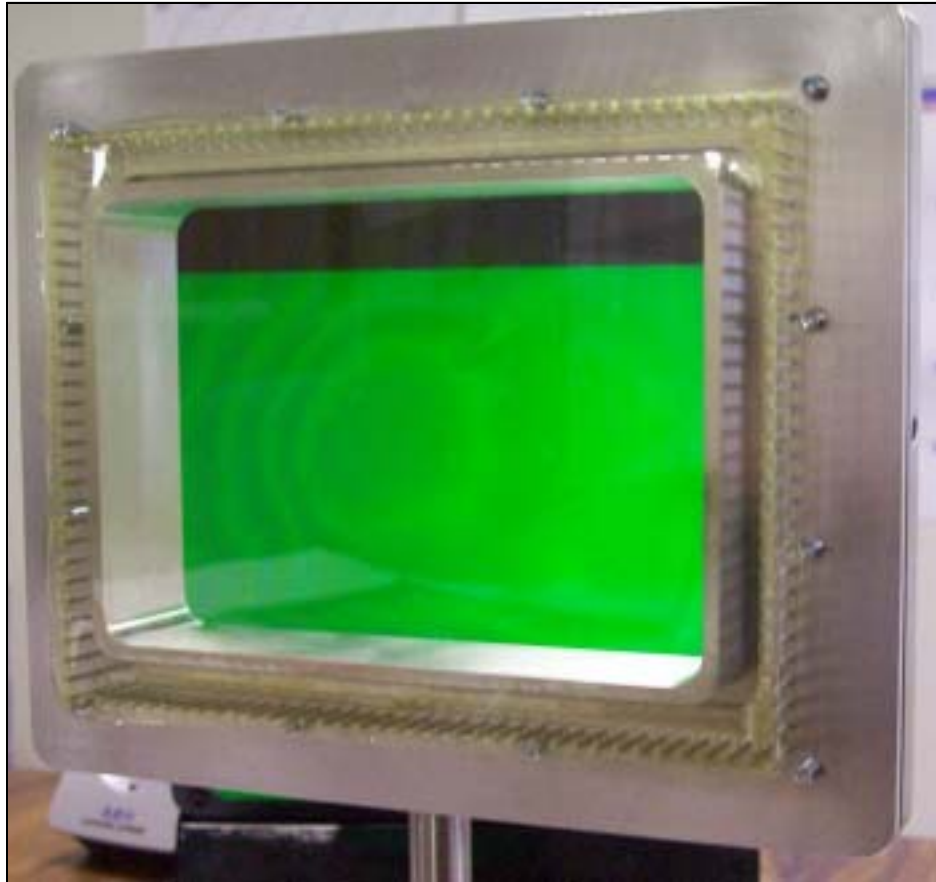


Figure 1.1: Rectangular pellicle beam splitter

Upon inspection, the rectangular form of the mount causes undesirable fringe patterns in the corners. The purpose of this project was to understand and reduce this undesirable optical response. The objectives were as follows:

1. Explain the physical reasons for corner fringe patterns
2. Use analysis to reduce the corner effects
3. Provide guidance in prototype design

As the ability to analyze optical data increases with technology, so does the need to use better performing optics. With the high cost of manufacturing and testing quality optics, it becomes important for engineers to rely on analysis to predict optical quality under different mechanical and thermal loads. One efficient way to analyze optics is to use finite element analysis. Finite element analysis (FEA) software is used to model the structure, apply loads, and view the effects of the loads on the structure. This application makes it possible to modify the structure and/or loads until desirable results are achieved. Creating models that accurately represent the optic is key to successfully understanding and mitigating undesirable optical performance.

Arnold Engineering Development Center

There are two cryogenic vacuum chambers, 7V and 10V, used at Arnold Engineering Development Center (AEDC), part of Arnold Air Force Base, Tullahoma, Tennessee. They are used to provide optical component testing and evaluation for the DOD, NASA, and other aerospace programs. These chambers are used to calibrate sensors, simulate missions, and simulate hardware-in-the-loop, all in low infrared. The 7V chamber, Figure 1.2, is used for calibration and characterization, while the 10V chamber is used for mission simulation. The chambers operate at temperatures as low as 20 K and pressures as low as 10^{-6} torr [Lowry *et al.*, 2005a; Lowry *et al.*, 2005b].



Figure 1.2: 7V Optical bench (Courtesy of AEDC)

The chambers are filled with many optical components including beam splitters, beam combiners, mirrors, attenuators, and other components (Figure 1.3). Most components of the optical chambers, including the optical bench, are made of aluminum. Aluminum is used for all components so that the coefficient of thermal expansion (CTE) is the same and everything stays in focus while cooling down to cryogenic temperatures. The large number of components leads to long optical paths. Each optical component induces some error into the system; therefore it is necessary to design each component for optimal performance.

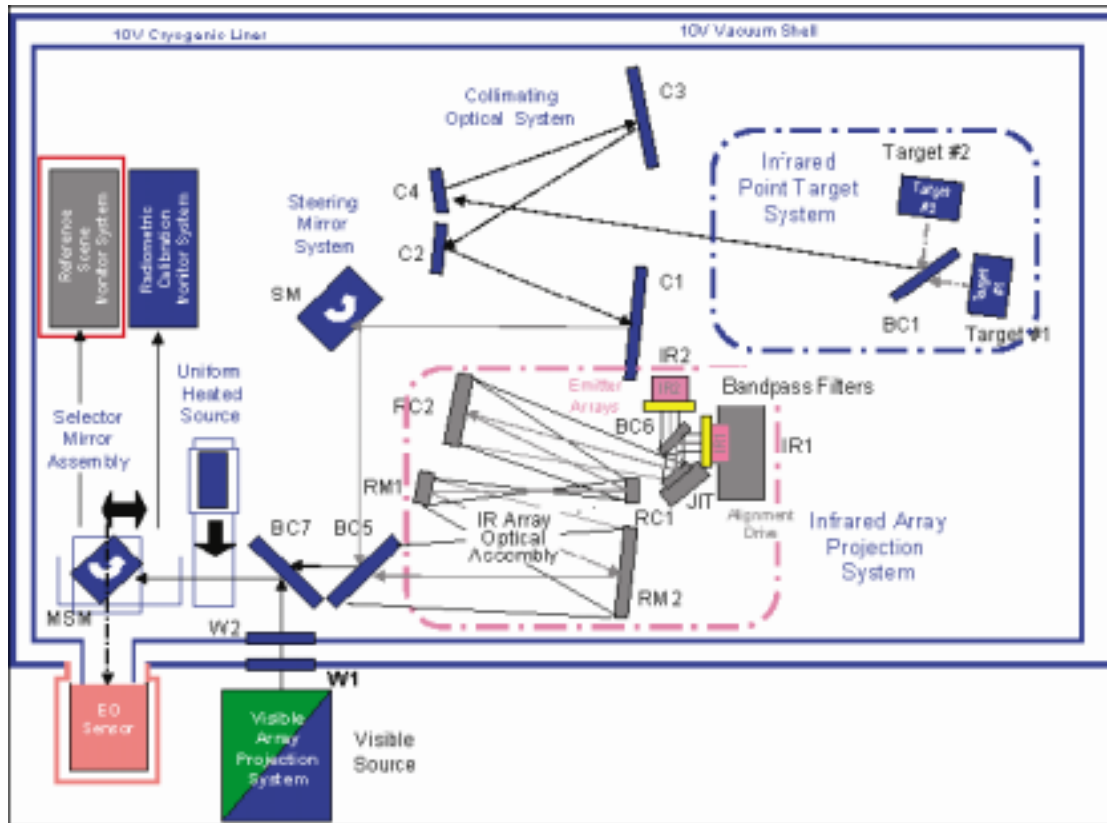


Figure 1.3: 10V Chamber optical configuration (Courtesy of AEDC)

Pellicle Beam Splitter

Beam splitters are one of the many types of components used in optical chambers. A beam splitter is an optical device that takes an incoming image and divides it into two paths by transmitting part of the image in one direction and reflecting the rest in another direction. An optical coating applied to the substrate controls the amount of the image that is transmitted or reflected. The two paths can be used to filter or analyze the image in different ways.

The proposed AEDC beam splitter is created by bonding a pre-stressed pellicle to an outer rectangular mount made of 6061-T6 aluminum (to match the other aluminum

components). An inner rectangular ring of aluminum is machined using a diamond turning process to produce an optical quality surface. This inner ring is then pressed against the pellicle to force it to be optically flat. A thin coating is applied to the pellicle to produce the required levels of transmission and reflectance. Figure 1.4 shows a pellicle beam splitter during optical quality testing.

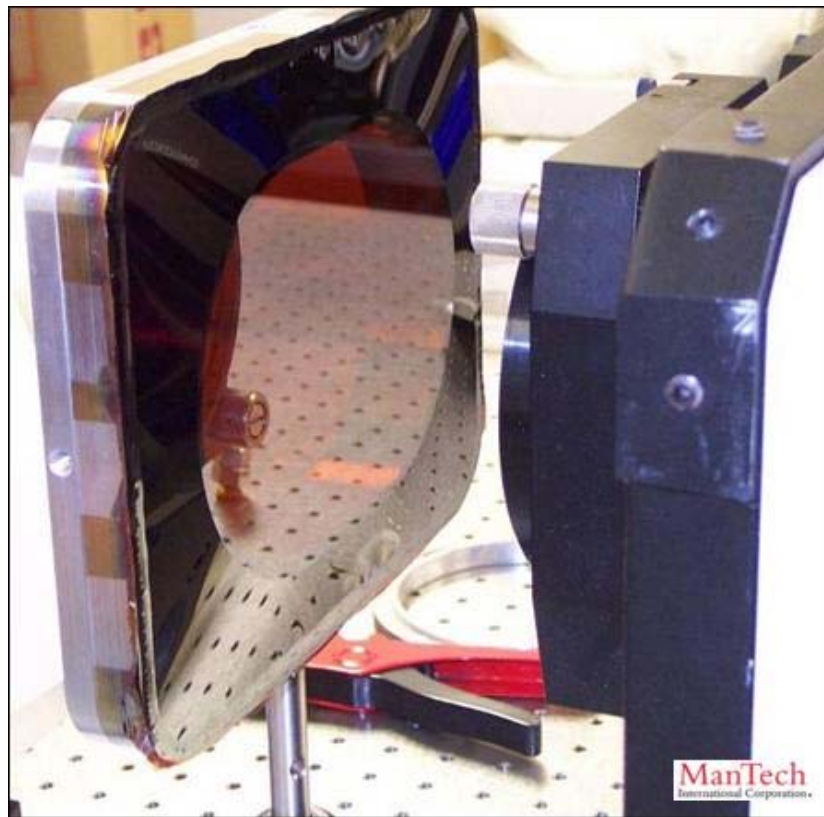


Figure 1.4: Pellicle beam splitter (Courtesy of GATR)

The use of a rectangular outer support and inner tensioning ring cause the pellicle to have a non uniform stress distribution. The corners of the rectangular frame result in fringe patterns that are undesirable in a precision optic. MSU was contracted to use finite element analysis (FEA) to study these corner effects.

Finite Element Analysis

Due to the high cost of manufacturing quality optics and testing them physically, it is desirable to use computer modeling to determine results beforehand. Finite element analysis (FEA) allows an in-depth look at stress and strain distributions. Changes can be made to the model quickly and efficiently. Many changes were made, including inner and outer ring dimensions, corner radii, pellicle thickness, and inner ring displacement. FEA was used to simulate the beam splitter's properties including dimensions, materials, and loads. FEA models are made with assumptions to simplify the analysis. These assumptions can have negligible effects on the results or can render the results unusable, making it important to have a strong understanding of FEA and to not completely trust the results without further verification.

LITERATURE REVIEW

The extent of optics technology is broad and deep but this does not mean it is complete. Demand for better performing optics is ever increasing, placing tighter standards on optical quality, weight, and durability. Some optics must be able to operate at the low temperatures and pressure of outer space. Meeting rigorous design parameters requires utilizing past knowledge and expanding on it. The literature reviewed here is limited to that which has a direct relation to the current work.

Optics

It is important to have a standard way of determining optical quality. There are two common ways to classify error between the actual optic and the ideal one: figure and roughness error. Figure error is defined as low frequency error while roughness is defined as high frequency error. Roughness error can be attributed to materials processing while figure error can be attributed to several aspects of the optic configuration. The overall optical quality can be determined by taking the root mean square (RMS) of the total error. This error is commonly read as waves of error with respect to the wavelength of light (λ) used [Marker *et al*, 2001]. This error is absolute over the entire optic and therefore not size dependant. A typical optical precision requirement is RMS figure error less than $\lambda/20$. Adding to this, the error of each optical component in the system is summed together (root mean square), making it increasingly complex to meet RMS error requirements as optical aperture increases.

Membrane optics have become more prevalent because there is a desire for larger yet lighter and cheaper optics. This desire is due to having limited launch volume and weight when sending satellites into space. The resolution of an optic directly correlates to its diameter [Blonk *et al*, 2006]. Traditional space optics are made of stiff and heavy materials such as glass that can be ground into the desired shape. The current largest space telescope is NASA's Hubble Space Telescope (HST) with a mirror diameter of 2.4 m. The telescope is made of glass that took many months to figure adequately. Making larger optics requires making them deployable in space since the cargo volume of the launch vehicle is limited. The James Web Space Telescope (JWST) is one example of a deployable optic. It has a planned deployable mirror of 6.5 m in diameter and an areal density an order of magnitude lower than the HST [Blonk *et al*, 2006]. While membranes are significantly lighter, there are challenges with structure stability and hence optical quality.

Rectangular Pellicles

Little work has been done in the area of rectangular pellicles, especially with an inner rectangular tensioning ring. A more common approach in designing pellicles is to use round boundaries that help generate the desired optical quality. There has been work in the area of pressurizing rectangular membranes. These pressurized rectangular membranes produce uneven stress distributions that radiate out from the corners. Stress equations derived for these pressurized membranes do not hold true as the corner is

approached [Otto, 1973], which shows the complicated corner effects associated with a rectangular membrane.

Coatings

Coatings are applied to optics to give the surface desirable properties. In a typical beam splitter, a coating is applied to obtain a 50% reflective and 50% transmissive surface. The reflective coating that is applied to the pellicle can lead to surface distortion or even failure. The problem in the coating arises in the mismatch of pellicle and coating coefficient of thermal expansion (CTE) and failure strain. This mismatch in CTE when combined with large temperature changes induces bending moments in the pellicle and will distort the surface figure. The coatings are often applied at high temperature and induce residual strains upon cooling, leading to further distortion of the pellicle surface figure. Under high thermal and/or mechanical loads, it is possible for the coating to crack or delaminate due to strain mismatch. CTE of the coating can vary depending on deposition temperature and must be carefully monitored [Soh *et al.*, 2005]. (The coating CTE mismatch was not the primary concern of the present work.)

Interferometry

One common way to measure optical performance is to use interferometry. Interferometry is the technique of superimposing (interfering) two or more waves to detect differences between them [Baldwin, 2001]. An interferometer works by splitting a wave into two or more paths with one path passing through, or reflecting off the optic.

The paths are then combined to create wave interference. Having a phase difference in any wave will cause interference; in an optic, this indicates surface figure error. Two waves in phase will add, while two waves that are 180 degrees out of phase will subtract. This wave interference measurement makes interferometry a sensitive measuring device [Baldwin, 2001].

PROTOTYPES

GATR Technologies in Huntsville Alabama has done prototype construction and testing for the pellicle beam splitter. In addition, they have done material testing to find a suitable substrate for the project. Testing of materials lead to a substrate choice of Novastrat Polyimide Series 070116.003 (NPS) because its CTE is similar to that of aluminum. Transmission, reflection, absorption, and modulus of elasticity properties were also tested. The prototype mounts were ¼ scale and machined out of aluminum. The inner ring surface was lapped to an optical quality finish while the outer mounting surface was of a common machine tolerance finish. A 50/50 reflective coating was applied to an 8.5 micron thick NPS membrane for all prototypes. The inner ring was pressed against the substrate by bolts around the perimeter as shown in Figure 3.1.

Small Corner Radius with Solid Outer Boundary

The first prototype beam splitter was fabricated with an inner ring finished to common machine tolerances. This led to so much figure error that the substrate was removed and the inner ring was lapped to optical quality. The substrate was then fixed to the outer boundary by means of an adhesive. A thin coating of aluminum was applied to the substrate to simulate a 50/50 reflective coating. The outer boundary was 152.4 mm (6 in) by 114.3 mm (4.5 in) with corner radii of 6.35 mm (0.25 in). Inner ring dimensions were 132 mm (5.2 in) by 94 mm (3.7 in) outside with a ring thickness of 2.54 mm (0.1 in) and corner radii of 6.35 mm (0.25 in). The small corner radius with a solid outer boundary prototype is shown in Figure 3.1.



Figure 3.1: Small corner radius prototype (inner ring visible) (GATR)

Small Corner Radius with Compliant Outer Boundary

In an effort to alleviate some of the uneven stress distribution in the pellicle, a prototype with a compliant outer boundary was constructed. It consisted of mounting the pellicle to an outer boundary of compliant pins by using a non-adhesion process. No reflective coating was applied to this prototype. There were 132 pins spaced evenly around the 152.4 mm (6 in) by 114.3 mm (4.5 in) outer boundary with no corner radii. The pins were aluminum with a diameter of 1.27 mm (0.05 in) and a height of 12.7 mm (0.5 in). Inner boundary dimensions were identical to the previous prototype. The prototype with a compliant boundary is shown in Figure 3.2.

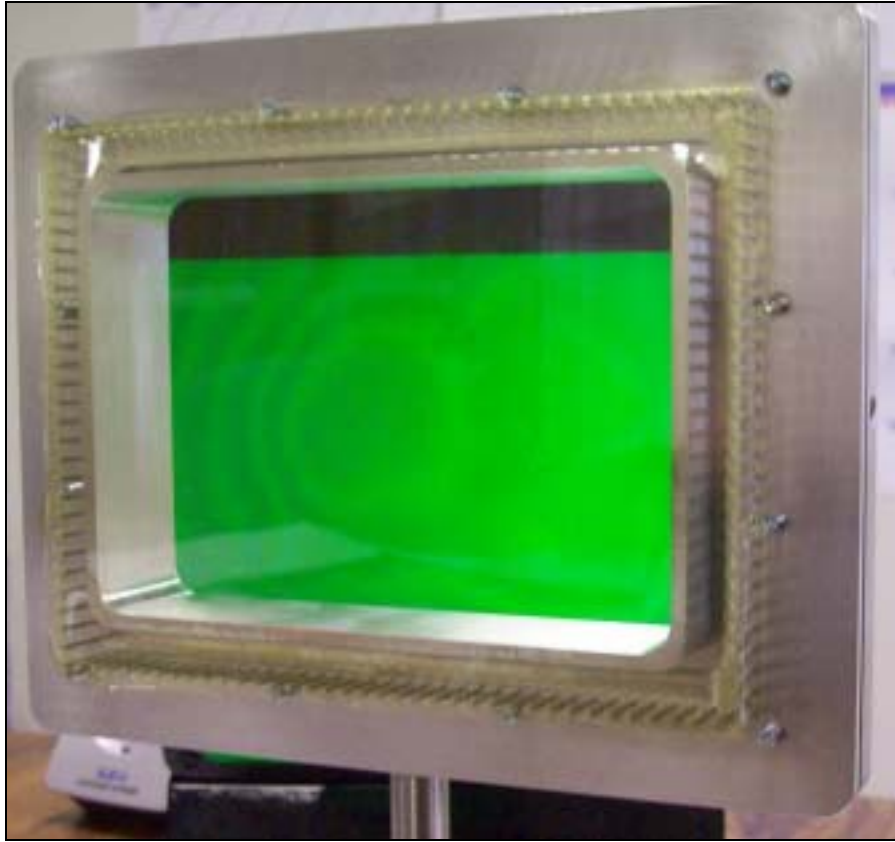


Figure 3.2: Small corner radius deformable boundary prototype
(inner and outer ring both visible)

Large Corner Radius Inner Ring

Using results from the FEA modeling, a final mount was made with large radius inner ring corners. A 50/50 reflective coating was applied to the substrate and bonded to the outer ring using a heat cure. The inner ring was lapped to optical quality and pressed against the substrate. The boundary dimensions were similar to the first prototype, but with large inner ring corner radii of 48.3 mm (1.9 in). Figure 3.3 shows the large corner radius prototype.

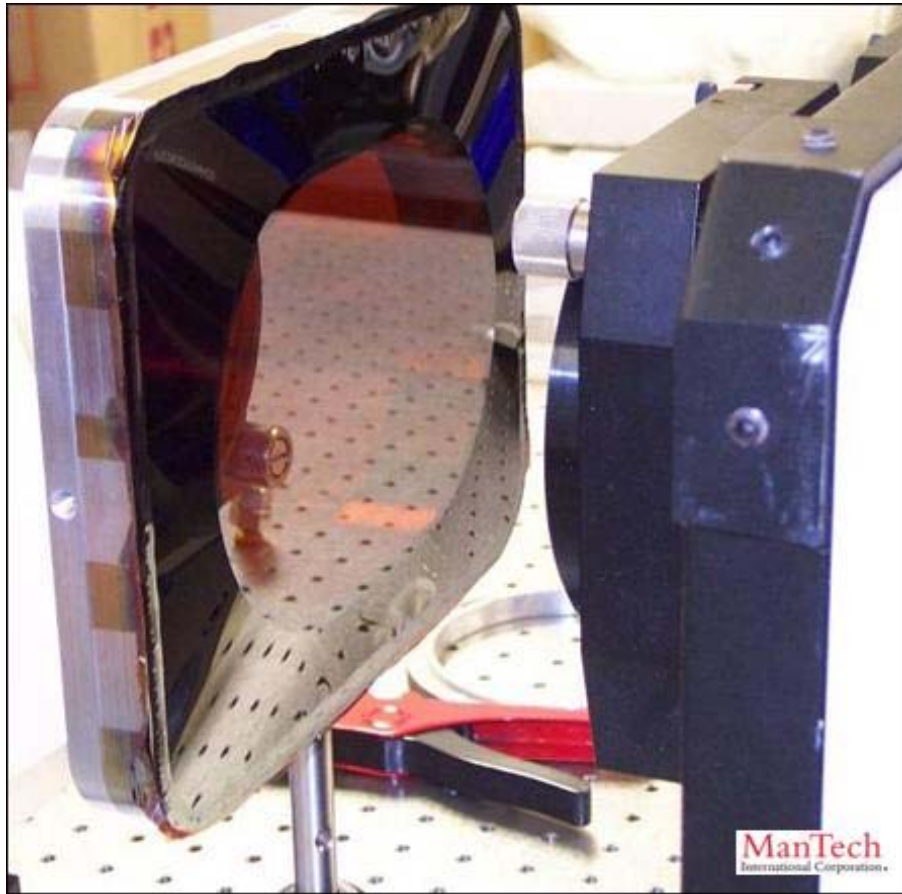


Figure 3.3: Large corner radius prototype

FINITE ELEMENT MODELS

General FEA Model Description

Modeling pellicle membranes in FEA requires an understanding of element types. Many element types are available in FEA for both solids and shells. The elements must be able to accept the desired loads while providing the desired results. For example, some solid elements are not able to represent thermal loads. More care must be taken when modeling a thin flexible structure like the pellicle. Membrane elements, a type of shell element, can only transmit “in-plane” forces [ABAQUS]. This means the membrane elements have no bending stiffness and can not resist moments. Shell elements allow bending stiffness and can provide through the thickness results. S4R shell elements were used to model the pellicle. C3D8R brick elements were used to model the mount. Both elements included reduced integration with hourglass controls to reduce computation time with minimal affect on accuracy [ABAQUS].

Element mesh density is important in creating models that run quickly and produce precise results. A mesh that is too coarse will allow the model to solve quickly but produce inaccurate results on complex models. Using too many elements will produce precise results at the cost of long computation time. At some point, as mesh density increases the solution will change a little while the computation time will increase significantly. At this point, a suitable mesh density can be determined. It is also important to have a higher mesh density in areas of importance and where gradients are high.

Boundary and load conditions are other important parts of FEA modeling. Any part of the model can be constrained in the degrees of freedom that include translation and rotation. These constraints can be applied to represent an axis of symmetry. This enables only half of a symmetric model with one axis of symmetry, or one quarter of a model with two axes of symmetry to be created. It is important to model the loads as accurately as possible, although it is not always possible to know exactly what they are. Loads can be applied instantly or over time. Contact between parts must be considered at the pellicle membrane to inner ring interface. This interface can be fixed (no slip), able to slide with a coefficient of friction, or be connected by springs. Simple models were made to investigate these interfaces and determine the best solution for the pellicle beam splitter models. For example, a model with pressure only was compared to a model with a displaced block with slip and the results were identical.

When deformations in a model become large, it is important to use non-linear geometry representations. For the pellicle beam splitter models, non-linearity was assumed if deformations exceed the pellicle thickness. Most models had inner ring displacements (2.54 mm) on the order of 100 times the pellicle thickness (0.0254 mm). This “large deformation” of the pellicle leads to solution convergence problems and was evident in initial model trials. To help insure solution convergence, ABAQUS non-linear solution capabilities were used. ABAQUS solves non-linear problems by slowly applying the load in steps (increments), where the previous step provides the initial state for the next step. This is repeated until the complete load has been applied. A Newton-Raphson technique was used to iterate to a solution within each step.

Many models were created to evaluate different affects on the rectangular pellicle surface figure. All parts of a model were kept constant except for varying one parameter at a time. Varying one parameter at a time allows for a better understanding of that parameter's effects. The parameters modified were as follows: pellicle coating, deformable outer boundary, pellicle thickness, inner ring displacement, catenary outer boundary, outer boundary corner radius, inner ring corner radius, and inner to outer boundary spacing. Figure 4.1 shows the FEA model parameters.

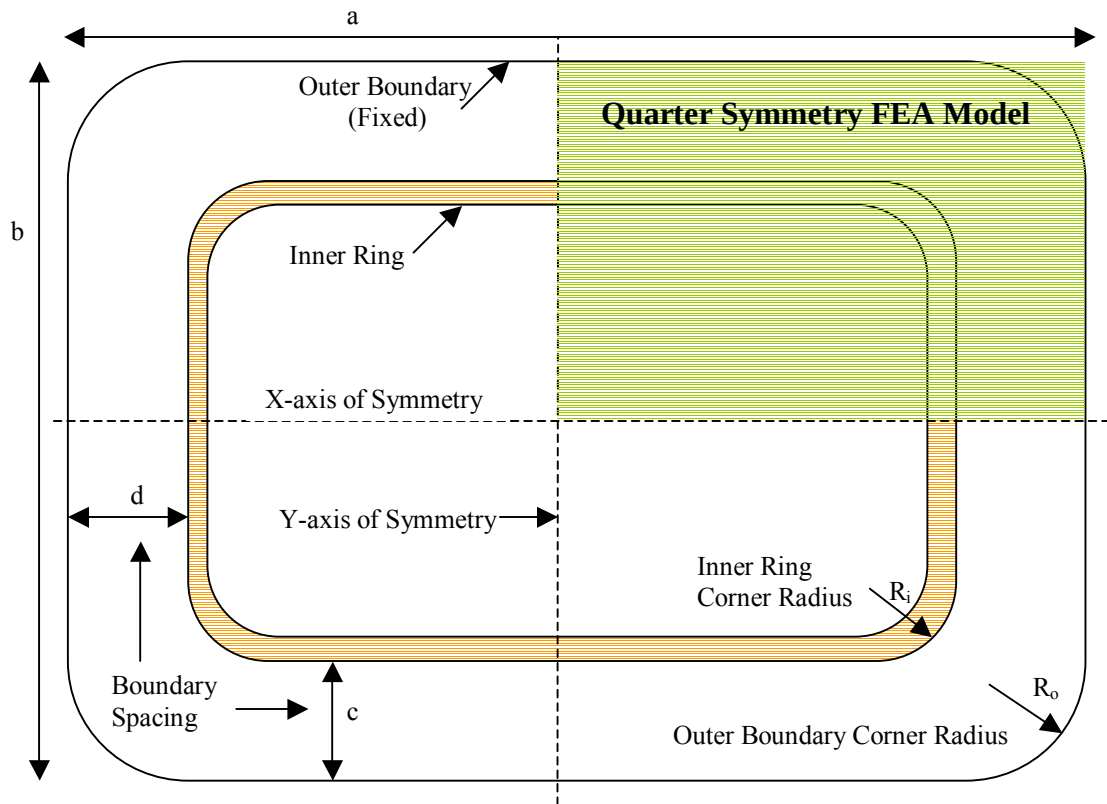


Figure 4.1: FEA model figure

All FEA models were created with a length to width ratio a/b of about $4/3$. Most models were $a=152$ mm (6 in) by $b=114$ mm (4.5 in). This is only one quarter scale of the required pellicle, but it matches the dimensions of the prototypes. In models where

the dimensions were changed, the 4/3 length to width ratio was kept constant. (This ratio was slightly varied at times to keep the modeling dimensions simple.) An inner ring displacement was used on most models. The models created were quarter symmetry models (symmetry about the global x and y axis – see Figure 4.1.1). Novastrat™ Polyimide Series (NPS) 070116.003 was used for most models of the pellicle and aluminum was used for the inner ring. The material properties of both materials are shown in Table 4.1. Coupon testing of NPS was shown to have an ultimate tensile strength of 131 MPa with the initial modulus being linear until around 80 MPa [GATR]. Most models had maximum stress values under 80 MPa, however, stress values are somewhat arbitrary to results. The Inner Ring Displacement Section in Chapter 5 shows how the results change very little when stress values are increased. High stress values could be avoided by simply displacing the inner ring less.

Table 4.1: Material properties (* estimate)

	Material	CTE (ppm/°C)	Modulus (GPa)	Poisson Ratio	Thickness (mm)
Substrate	NPS	24.6	4.57	0.4*	0.025
Ring	Aluminum	23.6	68.9	0.33	2.54

A mesh density convergence study was done. Based on this study, the mesh density of the models was set to 15.5 elements per cm² (100 per in²) for the pellicle and 61.0 elements per cm³ (1000 per in³) for the aluminum ring. This selected mesh density provided a reasonable compromise between solution accuracy and computation time. The interface between the inner ring and pellicle was modeled as frictionless because no friction value was available and this would allow the pellicle to slide freely over the ring as it was tensioned. Bonding the pellicle to the inner ring would not be a suitable option

since this would allow no change in the pellicle (inside the inner ring) as the inner ring was tensioned.

In most cases the outer aluminum support boundary was not modeled. Instead, the pellicle was modeled to the dimensions of the outer boundary and pinned at that location. Since most models did not have a temperature load applied to them it was assumed the outer boundary would not move and a pinned condition was a reasonable assumption.

A review of the typical model specifics is shown in the attached model input file (Appendix A). This input file is condensed to show element types, boundary conditions, and loads. Brief descriptions of important command lines are shown at the appropriate locations.

Pellicle Coating

The initial models were of a coated pellicle that was fixed to an outer aluminum support structure. There was no inner tensioning ring. CP1 was used as the substrate material for the pellicle and the coating was either CP1 or aluminum. The properties of CP1 are shown in Table 4.2. The model mesh and element cross section are shown in Figure 4.2 a and b.

Table 4.2: CP1 material properties (* estimate)

	Material	CTE (ppm/°C)	Modulus (GPa)	Poisson Ratio	Thickness (mm)
Substrate	CP1	51.2	2.17	0.4*	0.025
Ring	Aluminum	23.6	68.9	0.33	2.54
Coating	CP1/Al	51.2/23.6	2.17/68.9	0.4*/0.33	0.00025-0.0025

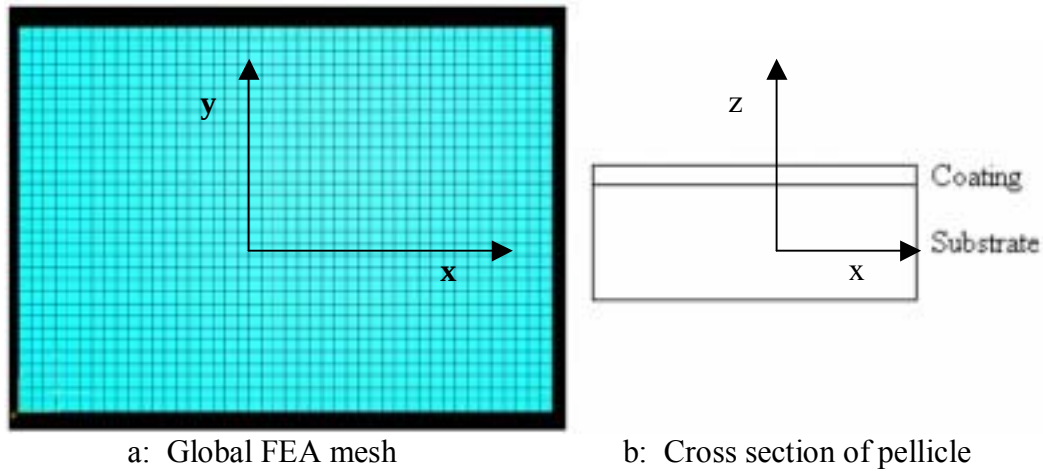


Figure 4.2: Pellicle coating FEA model

Simple models were made to develop baseline results, and to better understand and verify future models. A temperature change of $-200\text{ }^{\circ}\text{C}$ was applied to all models. (To ensure the boundary conditions and substrate properties were input correctly, a model was made with an applied pressure. This resulted in compressive edge effects that were similar to known solutions [Otto, 1973], providing confidence in the model.) Shell thickness was then varied to study its affect on the clear aperture. Also, one model was run with a temperature change of $-100\text{ }^{\circ}\text{C}$ and another with $-270\text{ }^{\circ}\text{C}$ to evaluate temperature range effects on the clear aperture.

Two models were constructed that considered the affect of the coating material. In one case the coating was considered to be the same material as the substrate (CP1). This ensured that the coating was modeled correctly, since the result should be the same as with no coating. In the other model the coating was considered to be aluminum. Aluminum was used since its properties notably differ from CP1, but is also a common coating material, and would give a good idea of the coatings affect on the model. The thickness of the coating was also varied to evaluate its affect on the pellicle.

Compliant Boundary

To alleviate some of the non-uniform stress in the pellicle, a prototype with a compliant outer boundary was constructed by GATR. It consisted of mounting the pellicle to an outer boundary of compliant aluminum pins. There were 132 pins evenly spaced around the 152 mm (6 in) by 114 mm (4.5 in) outer boundary. The pins were aluminum with a diameter of 1.27 mm (0.05 in) and a height of 12.7 mm (0.5 in).

To simplify the FEA modeling, springs with a stiffness equivalent to that of the aluminum pins were used. The equivalent spring stiffness (k) of a pin was calculated by using the equation in Figure 4.3, where E is modulus of elasticity, I is moment of inertia, and h is pin height. A model of a pin with an applied load was compared to a model of an equivalent spring with an equal load. Equal displacements ensured a correct equivalent spring. Small deformations were assumed and therefore a linear spring was used. Appendix B shows an abbreviated input file of the spring command along the outer boundary.

$$k = \frac{3 * E * I}{h^3}$$

Figure 4.3: Equivalent aluminum pin stiffness equation

FEA models were made using rectangular boundaries with rectangular inner rings in dimensions according to the quarter scale prototype. Two models were made: one with a rigid support boundary and one with a compliant boundary equivalent to the pin connections in the aluminum pin frame. The compliant boundary was connected to springs at locations according to the aluminum frame pins. The models were set to press

the inner ring against the pellicle and to have no friction between the inner ring and the pellicle. The inner ring was considered to be aluminum while the pellicle was assumed to be CP1.

Inner Ring Displacement

The inner ring was forced against the pellicle by displacing it perpendicular to and toward the pellicle surface. The amount of inner ring displacement can be varied greatly. The prototype inner rings were displaced with care to not induce so much stress that the pellicles or their coatings would fail. (Failure was noted when the substrate ripped or the coating cracked [GATR].) This resulted in displacements roughly 3.18 mm (0.125 in) to 6.35 mm (0.25 in) in the z-direction. To see the affect of inner ring displacement on non-uniform stress, five models were created where only the inner ring displacement was varied. The five inner ring displacement models are described in Table 4.3.

Table 4.3: Inner ring displacement model dimensions

Case	Outer Boundary (mm)			Inner Ring (mm)			Ring Displacement (mm)
	Length	Height	Radius	Length	Height	Radius	
RD-01	177.8	139.7	0	139.7	101.6	0	0.508
RD-02	177.8	139.7	0	139.7	101.6	0	1.27
RD-03	177.8	139.7	0	139.7	101.6	0	2.54
RD-04	177.8	139.7	0	139.7	101.6	0	3.81
RD-05	177.8	139.7	0	139.7	101.6	0	5.08

Pellicle Thickness

It is possible to manufacture pellicles of different thicknesses, so the affect of changing pellicle thickness was investigated by creating 10 models with all the same parameters except pellicle thickness. The thickness was varied from 0.0127 mm (0.0005

in) to 0.508 mm (0.02 in). The outer and inner boundaries of the models had no corner radii, and an inner ring displacement of 2.54 mm (0.1 in) was applied. The 10 FEA models created are shown in Table 4.4.

Table 4.4: Pellicle thickness model dimensions

Case	Outer Boundary (mm)			Inner Ring (mm)			Material Thickness (μm)
	Length	Height	Radius	Length	Height	Radius	
PT-01	177.8	139.7	0	139.7	101.6	0	12.7
PT-02	177.8	139.7	0	139.7	101.6	0	25.4
PT-03	177.8	139.7	0	139.7	101.6	0	38.1
PT-04	177.8	139.7	0	139.7	101.6	0	50.8
PT-05	177.8	139.7	0	139.7	101.6	0	101.6
PT-06	177.8	139.7	0	139.7	101.6	0	152.4
PT-07	177.8	139.7	0	139.7	101.6	0	203.2
PT-08	177.8	139.7	0	139.7	101.6	0	304.8
PT-09	177.8	139.7	0	139.7	101.6	0	406.4
PT-10	177.8	139.7	0	139.7	101.6	0	508

Outer Boundary Corner Radius

A common solution to alleviating non-uniform stress in membrane optics is to use a circular boundary. To make the rectangular membrane more like a circular membrane, nine FEA models of varying outer corner radii were created. To allow larger outer corner radii an inner corner radius of 12.7 mm (0.5 in) was used. All the models had the same inner and outer dimensions, ring displacement (2.54 mm), material thickness (0.0254 mm), and material properties. The FEA model dimensions are shown in Table 4.5 and a typical FEA model is shown in Figure 4.4. Larger corner radii were not possible since the outer boundary would overlap the inner ring.

Table 4.5: Outer boundary corner radius model dimensions

Case	Outer Boundary (mm)			Inner Ring (mm)		
	Length	Height	Radius	Length	Height	Radius
OR-01	165.1	127	0.0	139.7	101.6	12.7
OR-02	165.1	127	6.4	139.7	101.6	12.7
OR-03	165.1	127	12.7	139.7	101.6	12.7
OR-04	165.1	127	19.1	139.7	101.6	12.7
OR-05	165.1	127	25.4	139.7	101.6	12.7
OR-06	165.1	127	27.9	139.7	101.6	12.7
OR-07	165.1	127	33.0	139.7	101.6	12.7
OR-08	165.1	127	38.1	139.7	101.6	12.7
OR-09	165.1	127	43.2	139.7	101.6	12.7

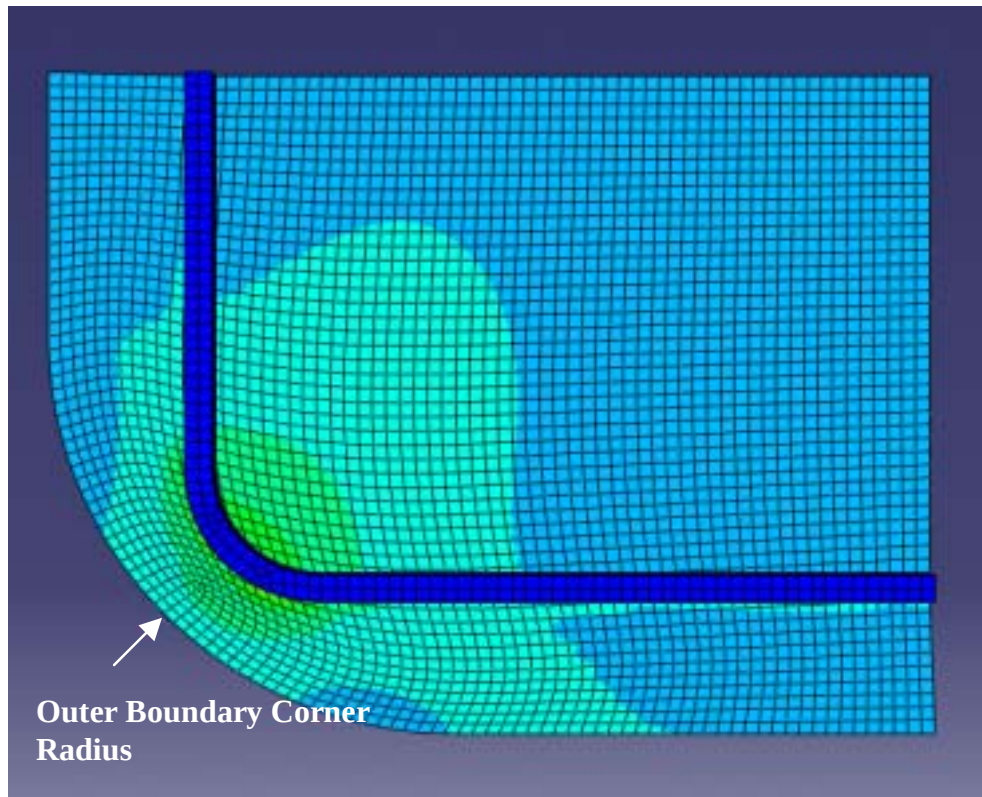


Figure 4.4: Typical outer boundary corner radius FEA model

Inner Ring Corner Radius

The effect of making the inner boundary more circular was examined similarly to that of the outer boundary. Eight FEA models were initially created to determine the

result of varying the inner ring corner radius. All other parameters of the models were kept constant. Viewing the results of the first eight models led to the need for further understanding and therefore two more sets of eight models were created. The only modification to each set was to the outer boundary dimensions. Table 4.6 shows the dimensions of the twenty-four models created. Figure 4.5 shows one of the FEA models.

Table 4.6: Inner boundary corner radius model dimensions

Case	Boundary Spacing (mm)	Outer Boundary (mm)			Inner Ring (mm)		
		Length	Height	Radius	Length	Height	Radius
IR-01	25.4	152.4	114.3	0	139.7	101.6	0
IR-02	25.4	152.4	114.3	0	139.7	101.6	5.08
IR-03	25.4	152.4	114.3	0	139.7	101.6	12.7
IR-04	25.4	152.4	114.3	0	139.7	101.6	25.4
IR-05	25.4	152.4	114.3	0	139.7	101.6	38.1
IR-06	25.4	152.4	114.3	0	139.7	101.6	43.18
IR-07	25.4	152.4	114.3	0	139.7	101.6	48.26
IR-08	25.4	152.4	114.3	0	139.7	101.6	50.8
IR-11	31.75	165.1	127	0	139.7	101.6	0
IR-12	31.75	165.1	127	0	139.7	101.6	5.08
IR-13	31.75	165.1	127	0	139.7	101.6	12.7
IR-14	31.75	165.1	127	0	139.7	101.6	25.4
IR-15	31.75	165.1	127	0	139.7	101.6	38.1
IR-16	31.75	165.1	127	0	139.7	101.6	43.18
IR-17	31.75	165.1	127	0	139.7	101.6	48.26
IR-18	31.75	165.1	127	0	139.7	101.6	50.8
IR-21	38.1	177.8	139.7	0	139.7	101.6	0
IR-22	38.1	177.8	139.7	0	139.7	101.6	5.08
IR-23	38.1	177.8	139.7	0	139.7	101.6	12.7
IR-24	38.1	177.8	139.7	0	139.7	101.6	25.4
IR-25	38.1	177.8	139.7	0	139.7	101.6	38.1
IR-26	38.1	177.8	139.7	0	139.7	101.6	43.18
IR-27	38.1	177.8	139.7	0	139.7	101.6	48.26
IR-28	38.1	177.8	139.7	0	139.7	101.6	50.8

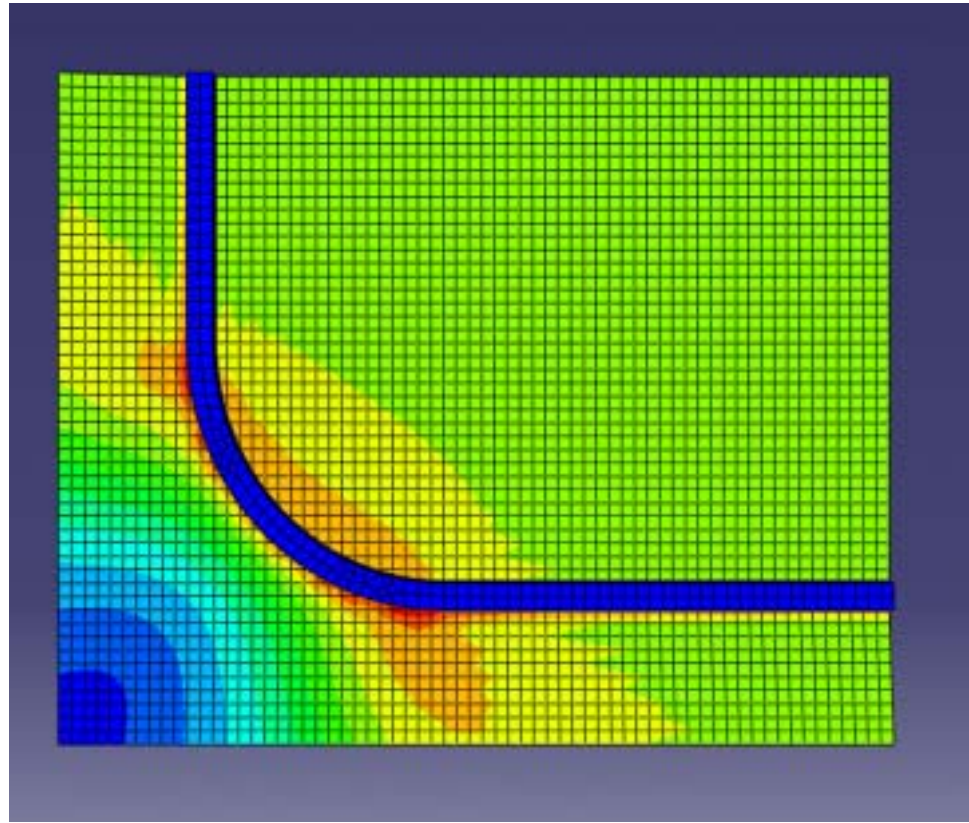


Figure 4.5: Typical inner ring corner radius FEA model

Inner Ring to Outer Boundary Spacing

In order to understand the relationship between inner and outer boundary spacing and non-uniform stress, a set of six models was created. The outer boundary dimensions were varied within each set and all other variables were kept constant. Once again, viewing the results for the first set led to a need for further understanding. Two more sets of models were created with each set having different inner ring dimensions but constant length to width ratios. The three sets of models dimensions are shown in Table 4.7 and a typical model is shown in Figure 4.6.

Table 4.7: Inner ring to outer boundary spacing model dimensions

Case	Boundary Spacing (mm)	Outer Boundary (mm)			Inner Ring (mm)		
		Length	Height	Radius	Length	Height	Radius
BS-01	6.4	111.8	83.8	0.0	99.1	71.1	0.0
BS-02	8.9	116.8	88.9	0.0	99.1	71.1	0.0
BS-03	15.2	129.5	101.6	0.0	99.1	71.1	0.0
BS-04	21.6	142.2	114.3	0.0	99.1	71.1	0.0
BS-05	34.3	167.6	139.7	0.0	99.1	71.1	0.0
BS-06	47.0	193.0	165.1	0.0	99.1	71.1	0.0
BS-11	6.4	147.3	109.2	0.0	134.6	96.5	0.0
BS-12	8.9	152.4	114.3	0.0	134.6	96.5	0.0
BS-13	15.2	165.1	127.0	0.0	134.6	96.5	0.0
BS-14	21.6	177.8	139.7	0.0	134.6	96.5	0.0
BS-15	34.3	203.2	165.1	0.0	134.6	96.5	0.0
BS-16	47.0	228.6	190.5	0.0	134.6	96.5	0.0
BS-21	6.3	182.9	134.6	0.0	170.2	121.9	0.0
BS-22	8.9	188.0	139.7	0.0	170.2	121.9	0.0
BS-23	15.2	200.7	152.4	0.0	170.2	121.9	0.0
BS-24	21.6	213.4	165.1	0.0	170.2	121.9	0.0
BS-25	34.3	238.8	190.5	0.0	170.2	121.9	0.0
BS-26	47.0	264.2	215.9	0.0	170.2	121.9	0.0

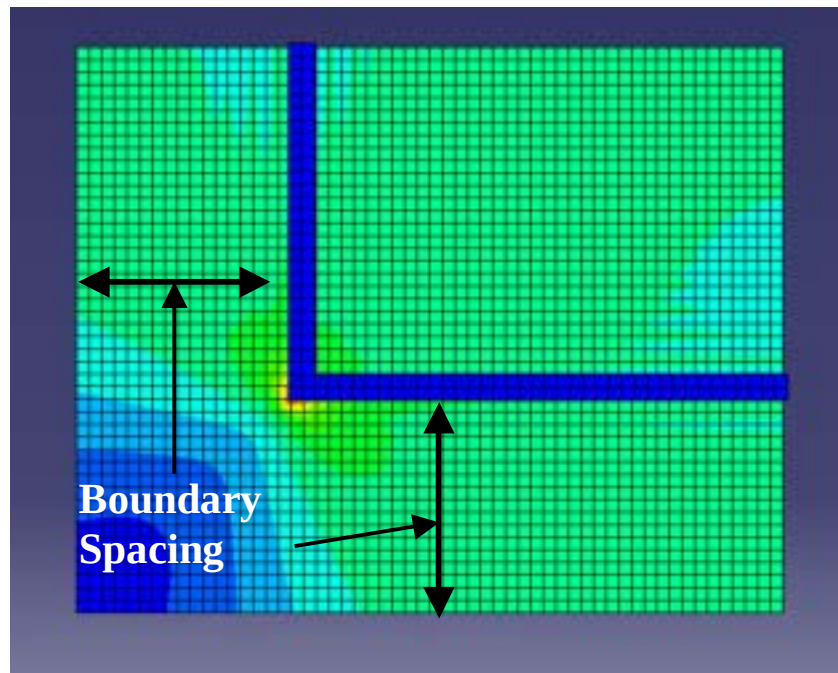


Figure 4.6: Typical boundary spacing FEA model

Catenary Outer Boundary

It was noted that a catenary boundary shape was used to help create a uniform stress distribution in a solar sail with deployable membrane sails [Greschick *et al*, 2003]. Originally, a strict triangular shape led to non-uniform stress and wrinkles. Making one side of the triangle into a catenary shape alleviated these problems by applying a uniform stress along the boundary. Using a catenary boundary shape was investigated in the pellicle beam splitter. Five FEA models were created with varying amounts of curvature in the outer boundary. The catenary shapes were parabolic and created arbitrarily at first. Using the first few models, a curvature was found that created a relatively even stress distribution along the inner boundary. An FEA model with an outer parabolic catenary boundary is shown in Figure 4.7.

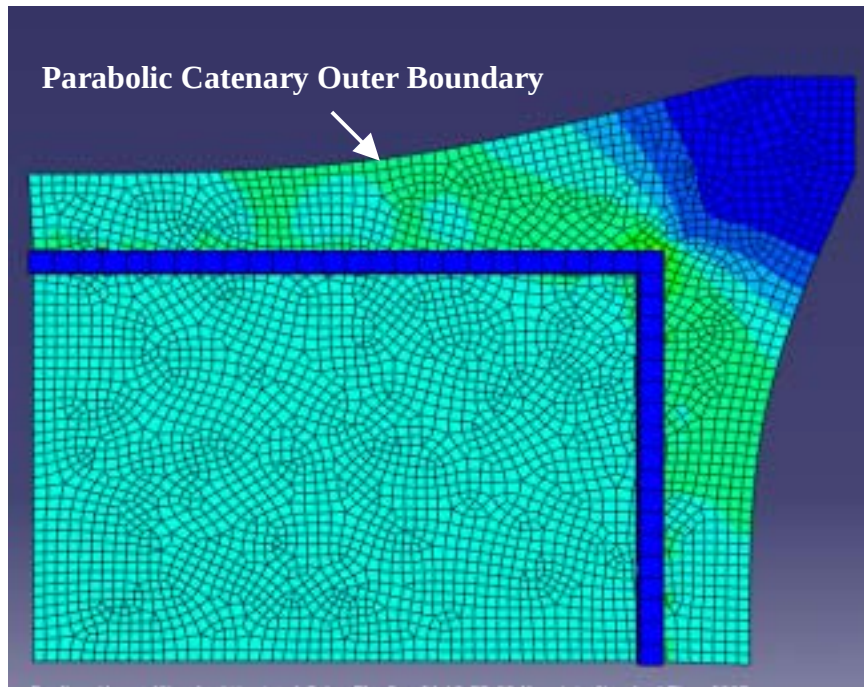


Figure 4.7: Typical catenary boundary FEA Model

One catenary model was also created with a large corner radius inner ring. The model was created by using the catenary boundary with the most uniform stress. This was combined with the inner ring corner radius that produced the most uniform stress distribution ($r = 48.3 \text{ mm}$) shown in Chapter 5. Figure 4.8 shows the FEA model with a parabolic catenary outer boundary and a large corner radius inner ring.

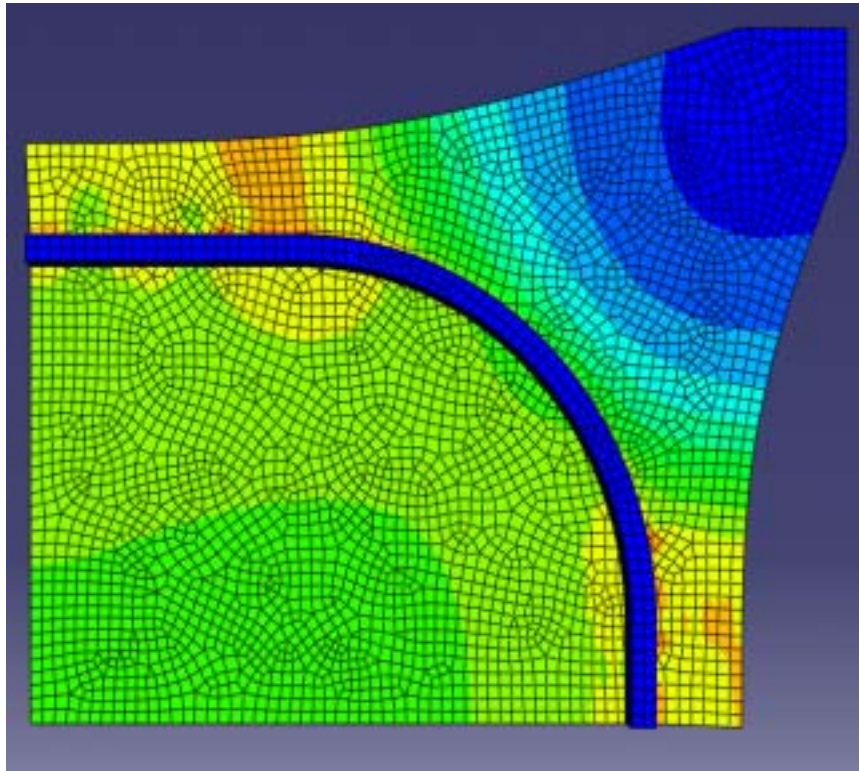


Figure 4.8: Catenary boundary with inner ring corner radius FEA model

RESULTS

Results Overview

The stress non-uniformity (η) of the surface figure was calculated by using FEA output values (at integration points) from inside the inner tensioning ring. The standard deviation of Von Mises (VM) stress distribution was divided by the average Von Mises stress value over the same area ($\eta = \text{VM standard deviation} / \text{VM average}$). The relationship of standard deviation to average is commonly called the coefficient of variation. Standard deviation was used to get a value for the variance of the data set from the mean. The average and standard deviation of a Gaussian function were used. (The distribution of the data was not confirmed to be Gaussian, but would have little affect on the results.) Von Mises stress was used since it is a common yield criterion. There were no negative minimum principle stresses and minimal shear stresses in most models, so the VM stress values were comparable to the maximum principle stresses. Most models showed maximum principle stress around 10 percent higher than maximum VM stress. Dividing the standard deviation by the average allows a way to normalize each case and compare them to one another. On models with radiused inner rings, a rectangular portion was chosen inside the inner radius, defining an effective aperture region. It was bounded by the x and y-axis of symmetry and point K (see Figure 5.1). This led to a 24% reduction in the effective aperture region on the largest inner ring radius models.

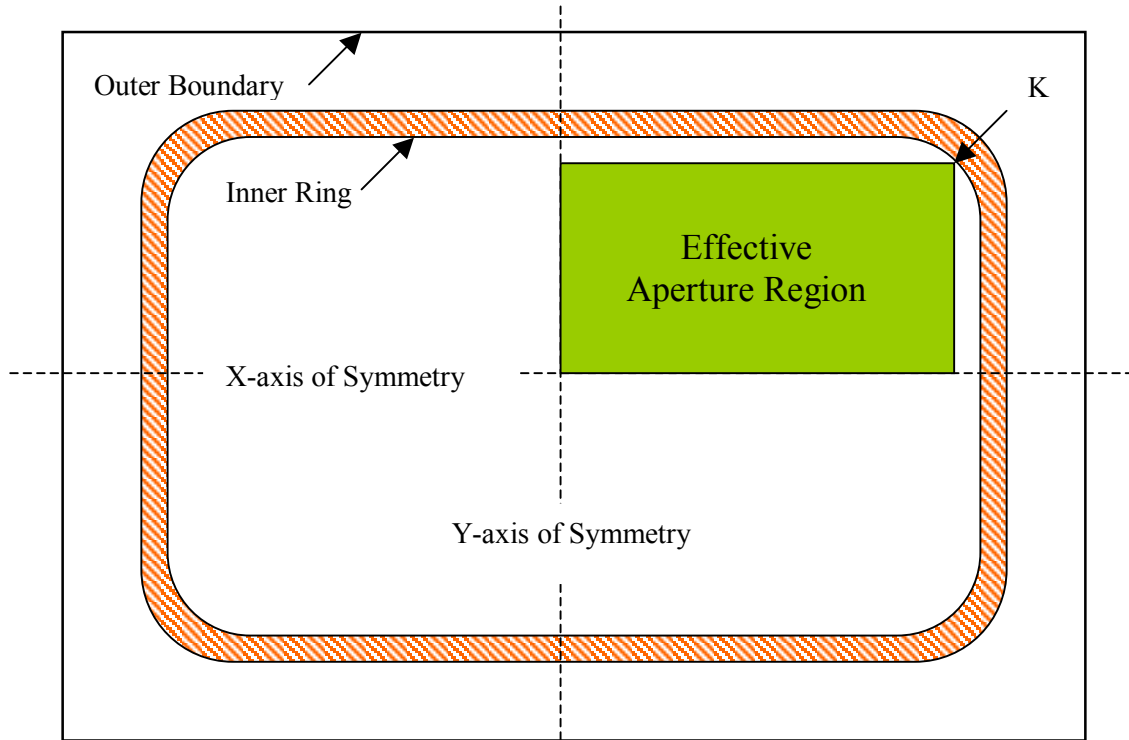


Figure 5.1: FEA model result output aperture

It is important to relate non-uniform stress to surface figure quality. Since there was no out-of-plane distortion, the change in membrane thickness was examined. Areas of non-uniform membrane thickness strain could result from non-uniform stress. For transmission optics, very small membrane thickness strains can lead to enough membrane thickness change to have undesirable effects on the surface figure. A study of the relation between membrane non-uniform in-plane stress and non-uniform thickness strain is shown in Appendix C. The conclusion was that in-plane non-uniform stress directly correlates to non-uniform thickness strain.

The mesh density of all models was kept the same. This creates many more elements in the aperture being studied as outer boundary size increases. A study was done to view the effects of changing mesh density on larger models to match the number

of elements on smaller models and is shown in Appendix D. The study showed that making the number of elements per model the same had similar results to keeping a constant mesh density.

Physical Explanation

It is important to understand what physically caused the corner fringe patterns. A simple one dimension FEA model was made to view the effect of moving a displaced block on membrane stress. The model, Figure 5.2, had a thin membrane with the left boundary pinned and all other boundaries bounded by symmetry. A solid block was displaced normal to the membrane (y-direction). Figure 5.3 shows the resulting maximum stress values as the block was moved in the x-direction starting at the pinned end.

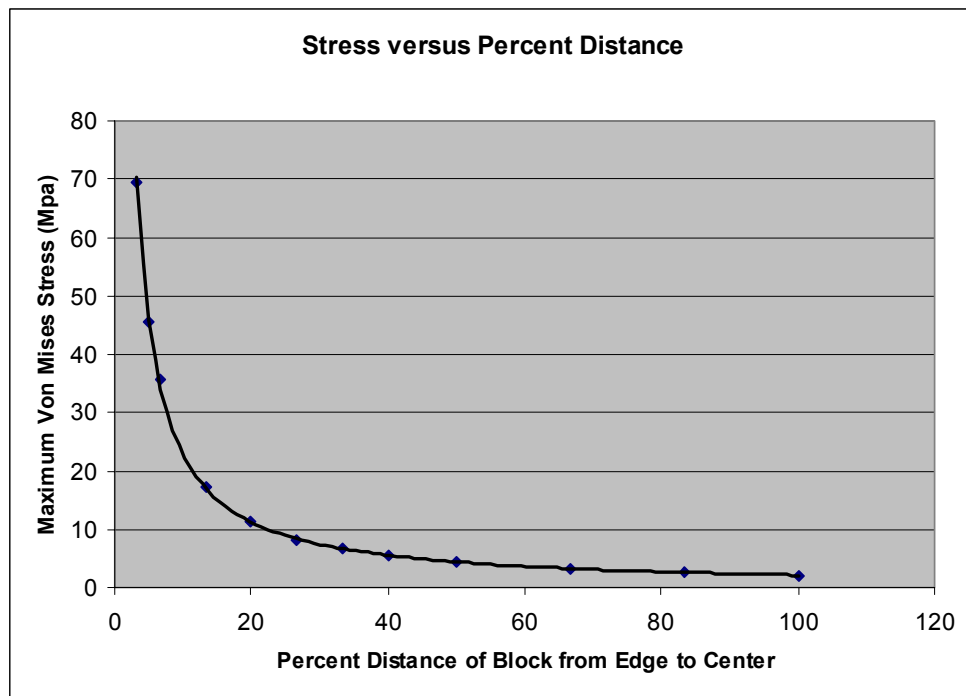


Figure 5.3: Plot of maximum stress versus distance from pinned edge

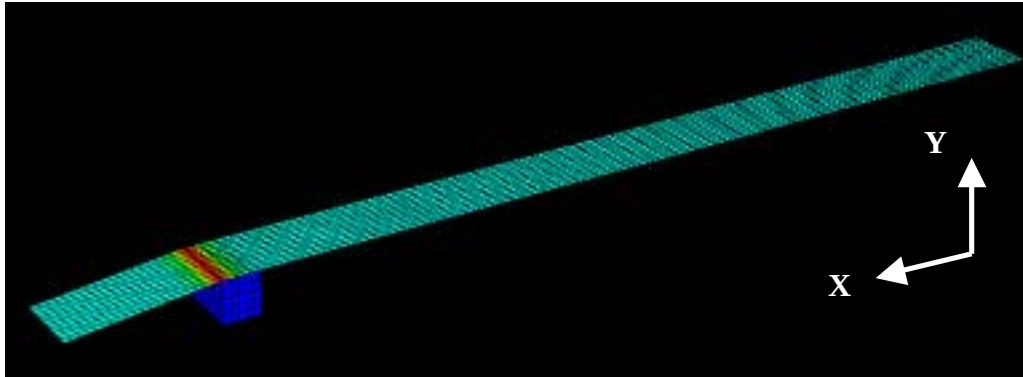


Figure 5.2: Simple FEA model for physical explanation

Figure 5.3 shows how stress decreases exponentially as the block is moved from the pinned end outward. The stress distribution of a square inflated membrane with a pinned outer boundary has known solutions. Figure 5.4 shows the stress distribution of an inflated square membrane [Otto, 1973]. The figure shows how non-uniform stress is developed in the corners and radiates toward the center. This non-uniform stress would lead to non-uniform thickness of the pellicle.

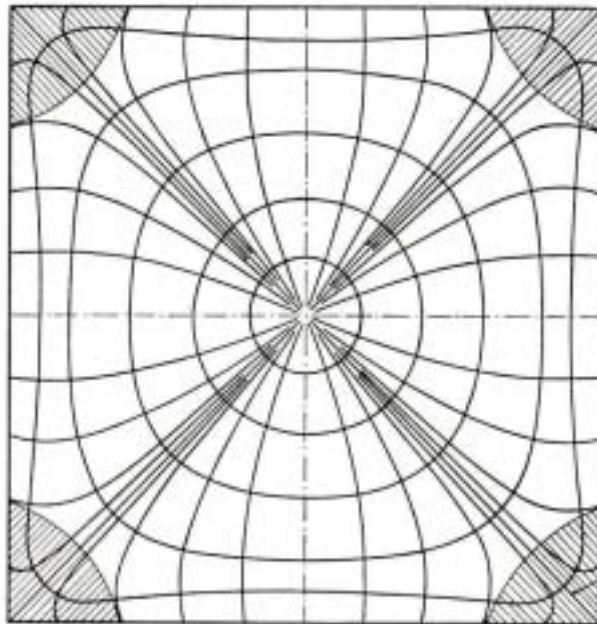


Figure 5.4: Stress contour plot of inflated square membrane [Otto, 1973]

Membrane optics are very sensitive to thickness change [Gierow]. Figure 5.5 shows how a wave front entering a varying thickness pellicle in phase will exit the pellicle out of phase. This happens because light waves travel slower through the pellicle material than air and therefore take longer to pass through. A pellicle of varying thickness will degrade optical quality. Thus, stress non-uniformity caused by the rectangular boundary creates non-uniform pellicle thickness and degrades optical quality

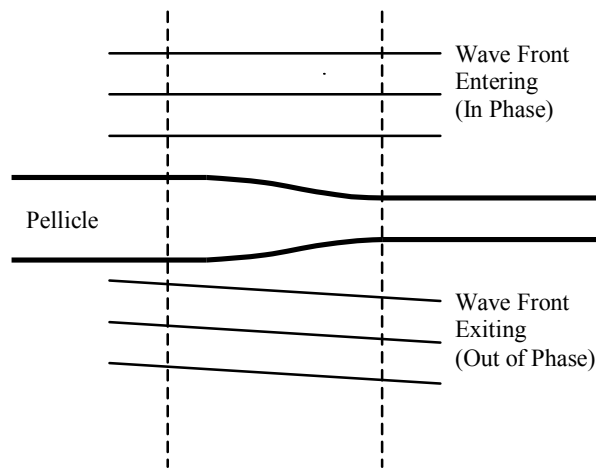


Figure 5.5: Pellicle thickness affect on optical performance

Pellicle Coating

Pellicle coating model details are described in Chapter 4. The trends for all models were identical. The only change between models was that the different temperatures applied affected stress values accordingly. Varying thickness or the number of elements did not affect surface quality. The results showed that principle stress was always uniform and never negative in the plane of the pellicle. Having only non-negative and uniform values of stress indicates that there should be no edge affects due to cooling the pellicle (with inextensible boundary). This can be seen in Figure 5.6 and 5.7.

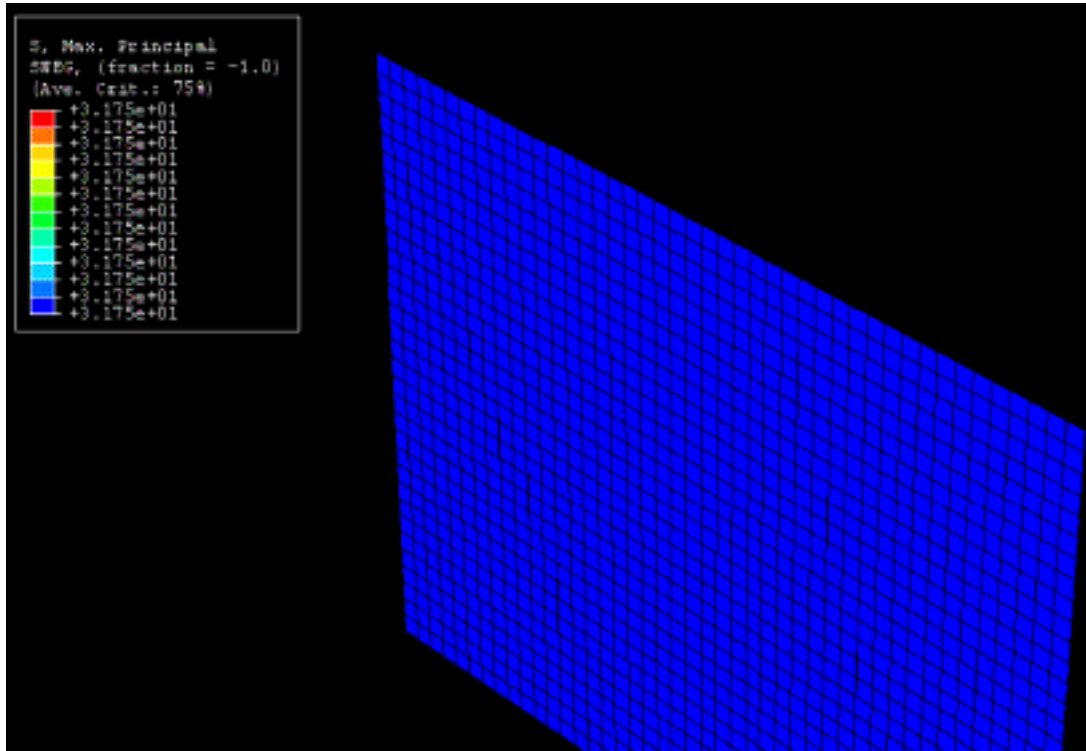


Figure 5.6: Typical pellicle maximum principle stresses

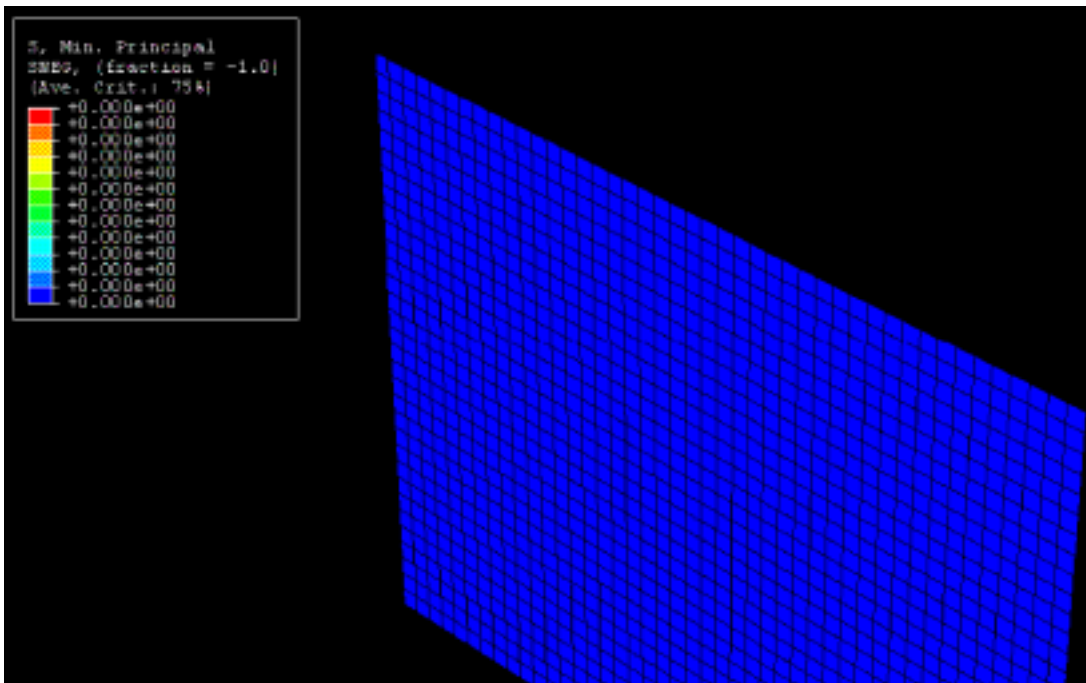


Figure 5.7: Typical pellicle minimum principle stresses

Compliant Boundary

Having a compliant outer boundary was investigated to help alleviate non-uniform stress and therefore increase surface quality. Compliant boundary model details are described in Chapter 4. The FEA results for models with a solid and a compliant outer boundary are shown in Table 5.1. Stress results of the two cases are shown in Figures 5.8 and 5.9. Maximum stress in both cases did not exceed 80 MPa.

Table 5.1: Compliant boundary results

	Von Mises Stress Values		
Outer Boundary	Average (MPa)	Std Dev (MPa)	η (%)
Solid	27.29	5.70	20.89%
Compliant	19.40	4.56	23.49%

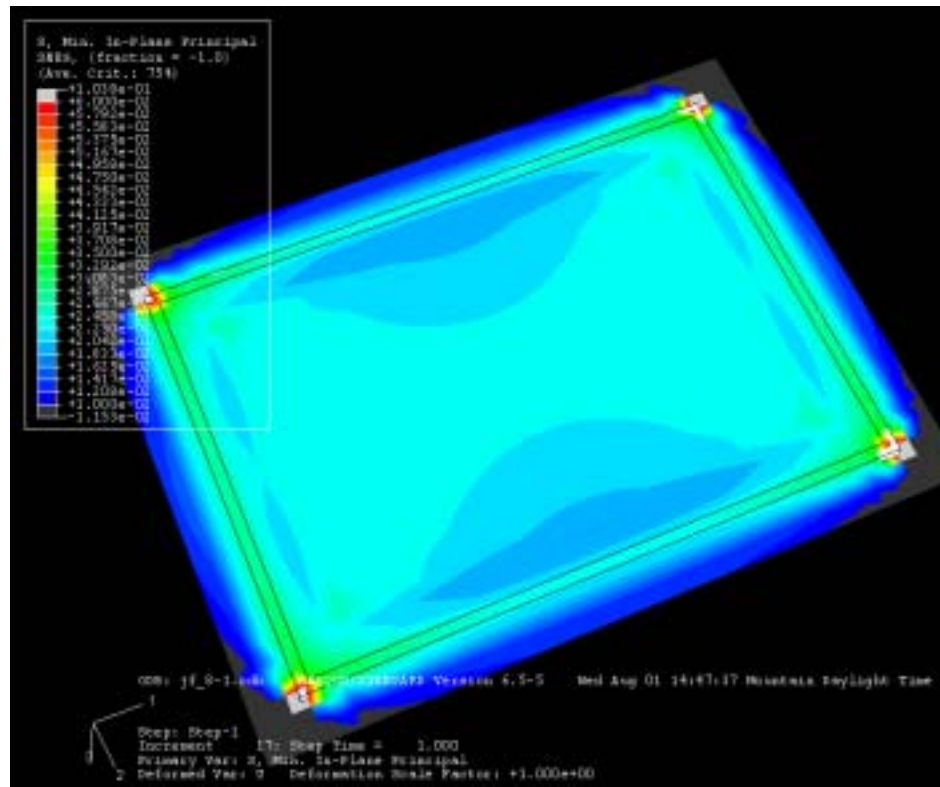


Figure 5.8: Minor (in-plane) principle stress (GPa) distribution with boundary rigidly supported

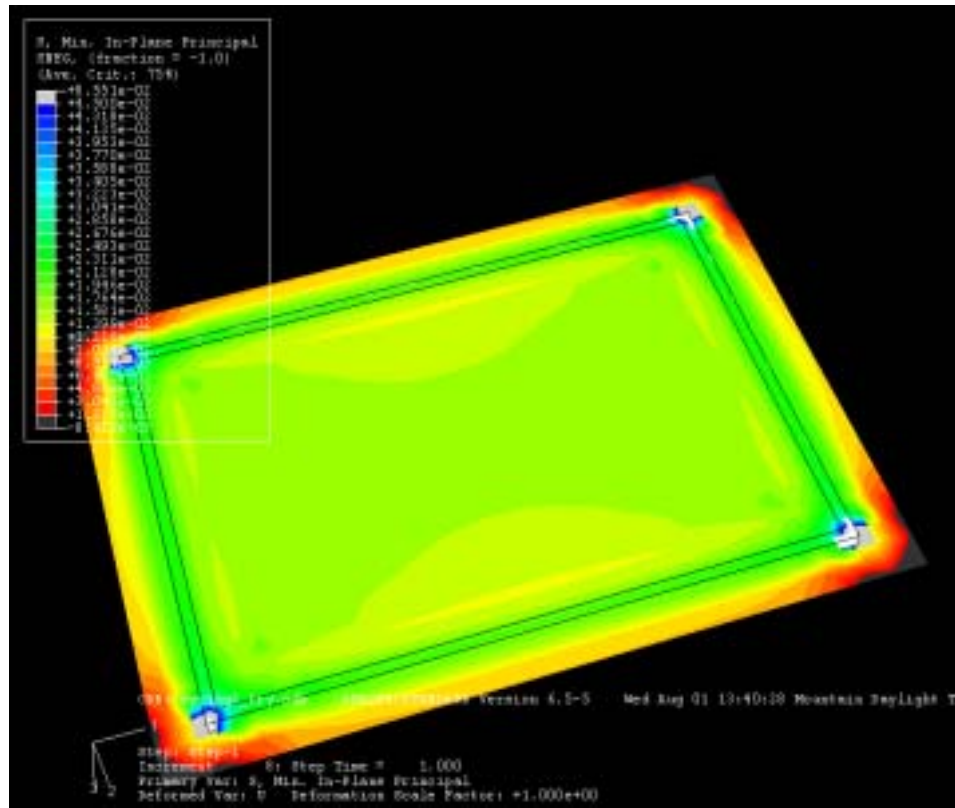


Figure 5.9: Minor (in-plane) principle stress distribution (GPa) with outer boundary supported by springs (notice about a 20% reduction in stress concentrations compared to Figure 5.4.1)

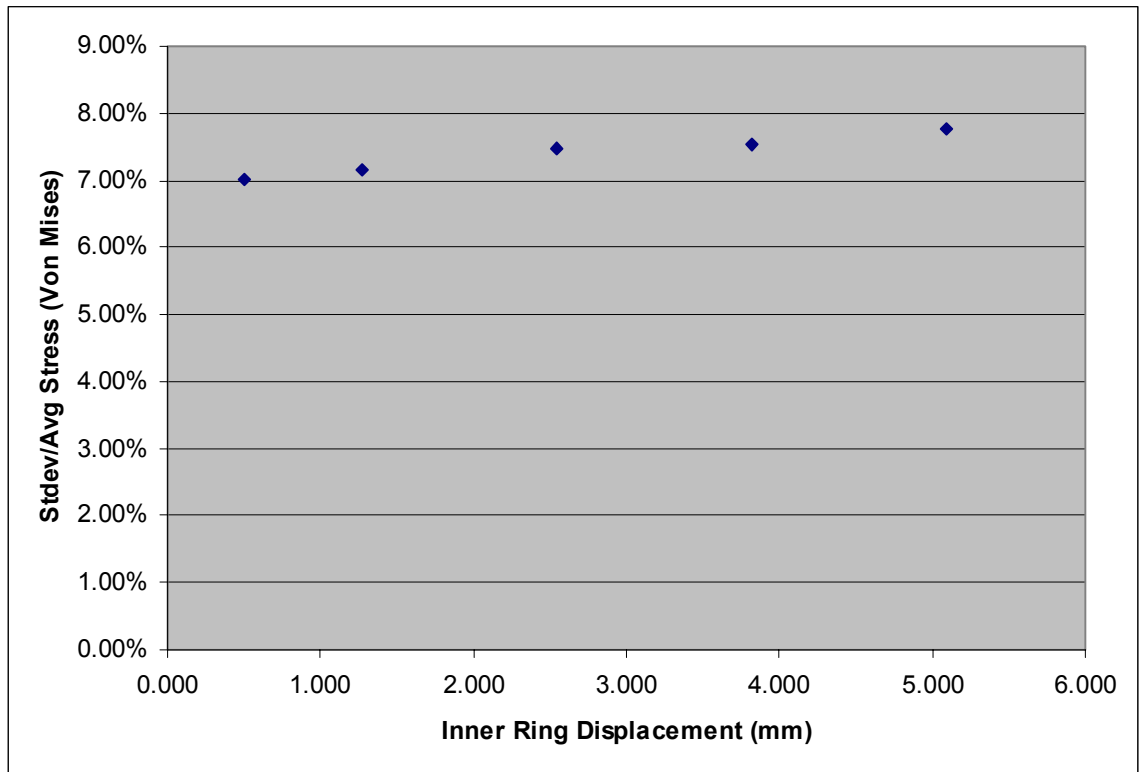
The two cases show important results. First, there is about a 20% reduction in stress concentrations when the compliant boundary is used. Also, the amount of non-uniform stress increased slightly with a compliant boundary and therefore does not increase pellicle surface quality.

Inner Ring Displacement

Inner ring displacement model details are described in Chapter 4. The five FEA model results for varying inner ring displacement are shown in Table 5.2 and a plot of these values versus inner ring displacement is shown in Figure 5.10.

Table 5.2: Inner ring displacement results

Case	Ring Displacement (mm)	Von Mises Stress Values		
		Average (MPa)	Std Dev (MPa)	η (%)
RD-01	0.509	0.6202	0.0435	7.01
RD-02	1.272	3.7526	0.2689	7.17
RD-03	2.544	14.8014	1.1079	7.49
RD-04	3.816	32.9836	2.4863	7.54
RD-05	5.088	58.1164	4.5136	7.77

Figure 5.10: Plot of stress non-uniformity, η , versus inner ring displacement

The amount of overall non-uniform stress is relatively insensitive to change as inner ring displacement is increased. Increasing the ring displacement four times only increases the amount of non-uniform stress 8.4%. While the amounts of non-uniform stress change slightly as the inner ring displacement is increased, the average amount of stress in the pellicle increases rapidly. With the same four times increase in ring displacement the average stress increases 14.5 times. This would quickly lead to

substrate and/or coating failure as the inner ring is displaced. Maximum Von Mises substrate stress for cases RD-01 to RD-03 was less than 50 MPa. Cases RD-04 and RD-05 had over 100 MPa maximum Von Mises stress leading to substrate yielding and possibly failure. A plot of substrate average stress versus inner ring displacement is shown in Figure 5.11.

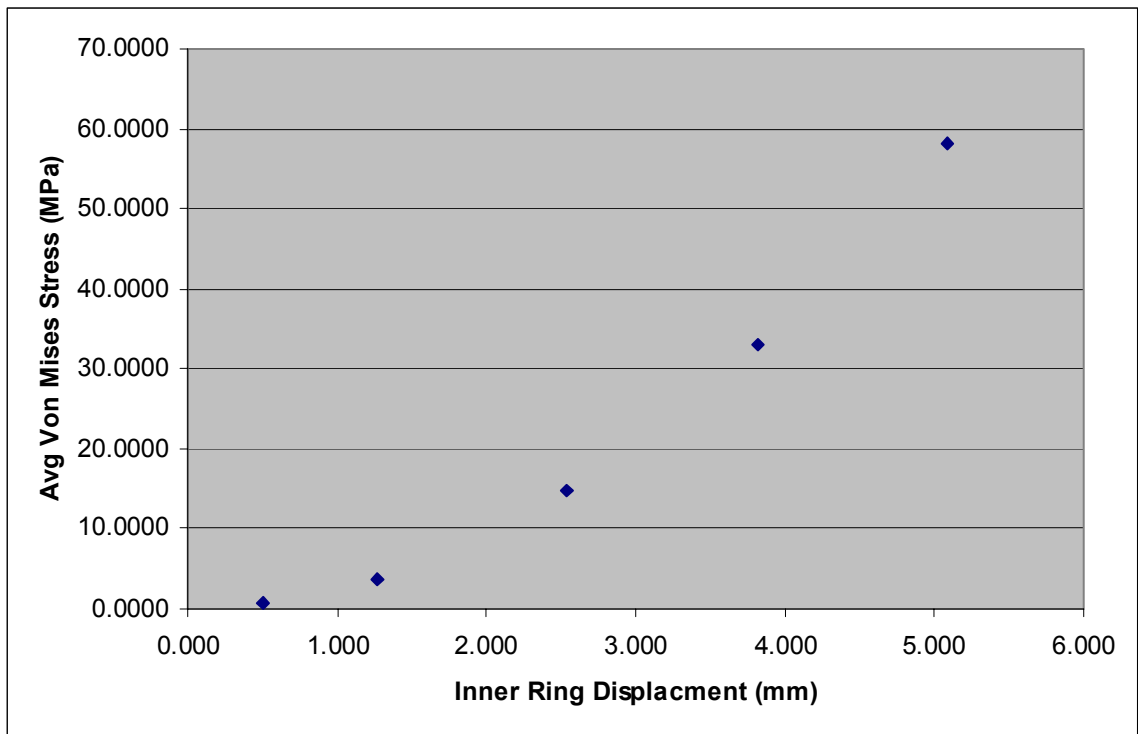


Figure 5.11: Plot of average Von Mises stress versus inner ring displacement

Pellicle Thickness

Pellicle thickness model details are described in Chapter 4. The average Von Mises stress and standard deviation of the stress were output from the ten FEA models with varying pellicle thickness. Then the standard deviation was divided by the average stress value for each case to provide normalized values for comparison. The ten models

and their respective values are shown in Table 5.3, and a plot of the values versus thickness is shown in Figure 5.12. Von Mises maximum substrate stress was over 90 MPa for cases PT-06 to PC-10 leading to possible yielding and even failure.

Table 5.3: Pellicle thickness results

Case	Material Thickness (mm)	Von Mises Stress Values		
		Average (MPa)	Std Dev (MPa)	η (%)
PT-01	0.0127	14.719	1.114	7.57
PT-02	0.0254	14.801	1.108	7.49
PT-03	0.0381	14.876	1.086	7.30
PT-04	0.0508	14.948	1.068	7.14
PT-05	0.1016	15.261	1.045	6.85
PT-06	0.1524	15.629	1.090	6.98
PT-07	0.2032	16.047	1.232	7.68
PT-08	0.3048	17.005	1.791	10.53
PT-09	0.4064	18.080	2.536	14.03
PT-10	0.508	19.242	3.340	17.36

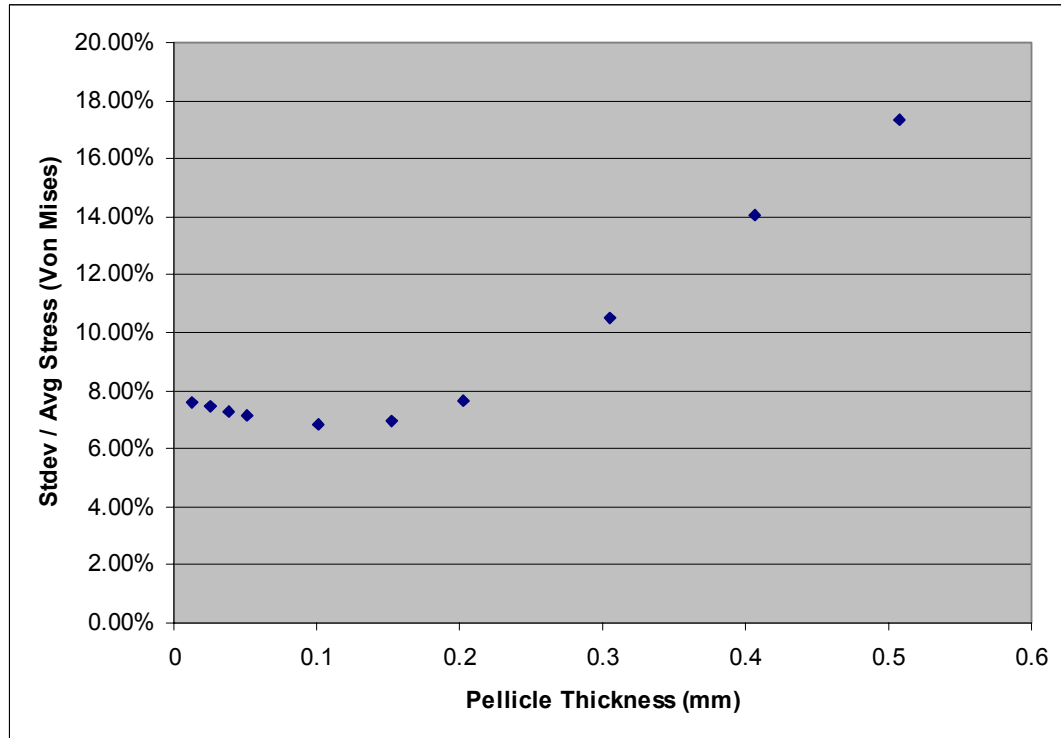


Figure 5.12: Plot of stress non-uniformity versus pellicle thickness

Figure 5.12 shows that the non-uniform stress initially decreases as pellicle thickness increases. Then at a thickness of 0.15 mm the amount of non-uniform stress increases in an almost linear manner. One possible explanation for this trend is that after a certain thickness the substrate's bending moment becomes significant enough to increase in-plane stress values. This would be due to the substrate bending over the inner ring and causing higher stress values at the inner ring boundary.

Outer Boundary Corner Radius

Outer boundary corner radius model details are described in Chapter 4. Results of the nine FEA models investigating outer boundary corner radius are shown in Table 5.4 and plotted in Figure 5.13. R^* column in Table 5.4 is calculated by taking the outer radius (R_o) and dividing it by the summation of the inner corner radius (R_i) and boundary spacing (D): $R^* = R_o / (R_i + D)$. R^* turns out to be an important parameter in the models. For these models $R_i + D$ was 25.4 mm (1 in), $R_i = 12.7$ mm (0.5 in) and $D = 12.7$ mm (0.5 in) (see Figure 5.14).

Table 5.4: Outer boundary corner radius results

Case	Outer Radius (mm)	R^*	Von Mises Stress Values		
			Average (MPa)	Std Dev (MPa)	η (%)
OR-01	0	0.00	24.09	1.691	7.02
OR-02	6.35	0.25	24.53	1.699	6.93
OR-03	12.7	0.50	24.56	1.764	7.19
OR-04	19.05	0.75	24.63	1.739	7.06
OR-05	25.4	1.00	25.00	1.843	7.37
OR-06	27.94	1.10	25.32	2.004	7.91
OR-07	33.02	1.30	26.41	2.524	9.56
OR-08	38.1	1.50	28.33	3.466	12.23
OR-09	43.18	1.70	32.62	6.782	20.79

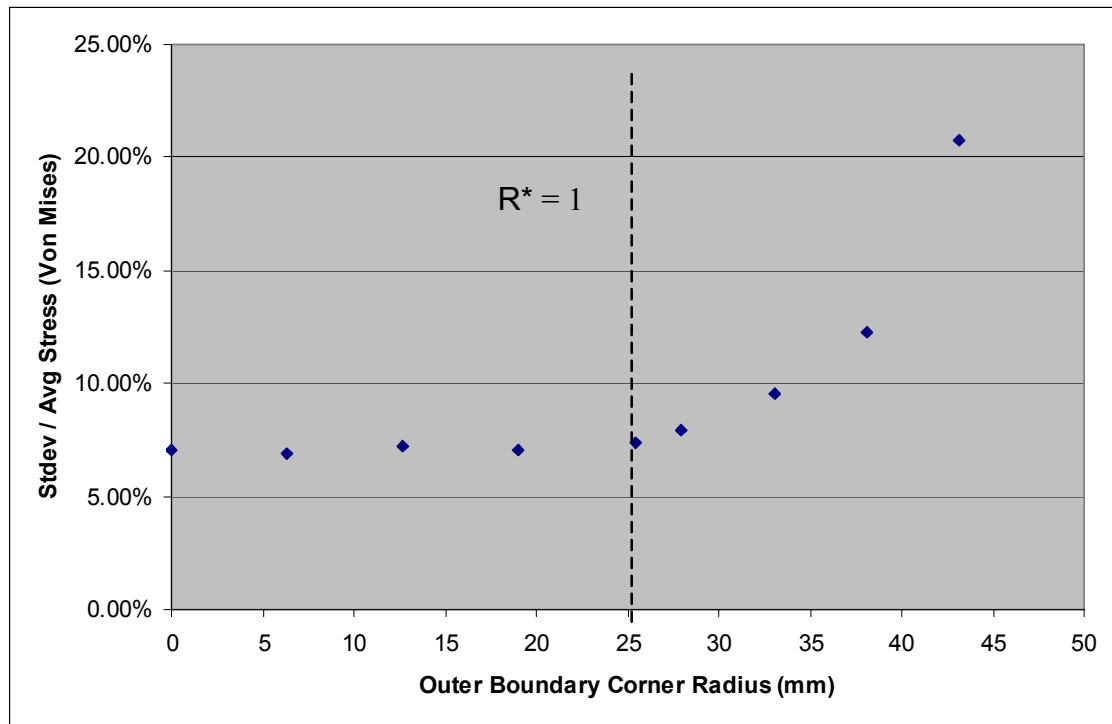


Figure 5.13: Plot of stress non-uniformity versus outer boundary corner radius

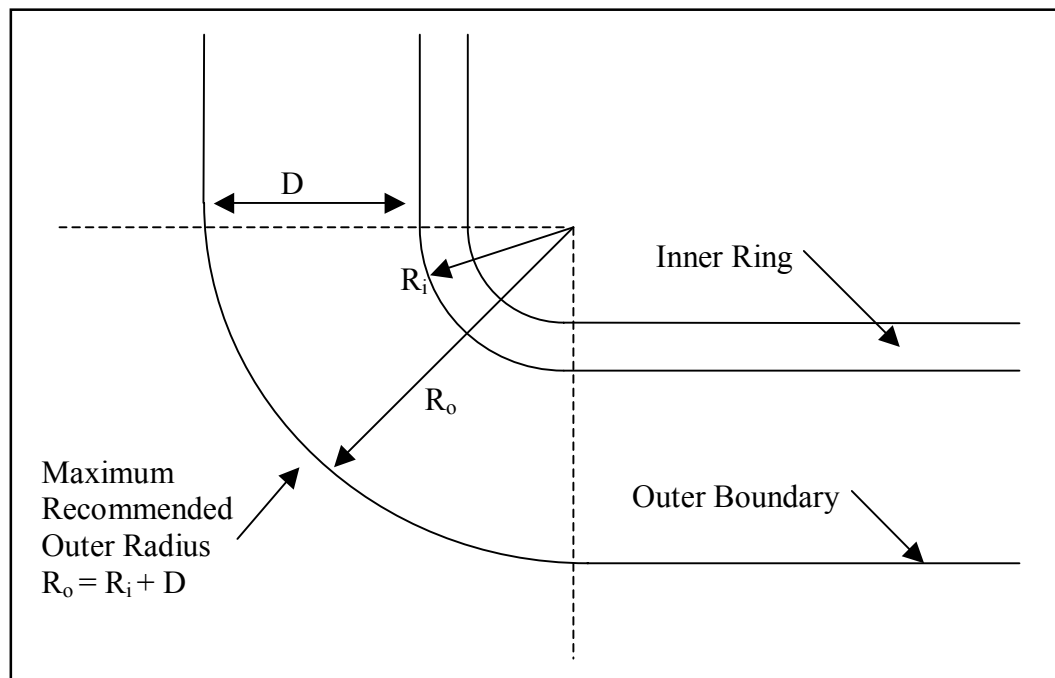


Figure 5.14: Maximum recommended outer boundary radius

Figure 5.13 shows that the amount of non-uniform stress stays relatively constant until R^* reaches a value of 1. Then the amount of non-uniform stress starts climbing at an exponential rate as R_o increases. Figure 5.14 shows the maximum recommended outer boundary radius to not adversely affect non-uniform stress. As R_o is increased past $R_i + D$ the amount of available material at the edges becomes reduced enough to create stronger stress concentrations at the corner and increase the amount of non-uniform stress. Cases OR-01 to OR-07 had Von Mises substrate stress values under 60 MPa while cases OR-08 to OR-09 had values over 80 MPa.

Inner Ring Corner Radius

Inner ring corner radius model details are described in Chapter 4. Upon viewing the results of the first eight FEA models, it was apparent that more models need to be created to better understand the affect of varying inner ring corner radius (R_i) on non-uniform stress. Two more sets of models were created with increased boundary spacing (D). Results of the 24 FEA models are shown in Table 5.5 and are plotted in Figure 5.15. Cases IR-01 to IR-04 and IR-11 all had maximum Von Mises stresses of over 80 MPa, leading to possible substrate yielding.

Figure 5.15 shows how the amount of non-uniform stress initially declines as inner ring corner radius is increased. This reduction can be significant as seen by comparing case IR-01 to IR-07 where non-uniform stress is reduced almost four times. At large radii, the amount of non-uniform stress levels off and then increases at the maximum radius. In this case, the maximum radius was 50.8 mm (2 in) since the width

was 101.6 mm (4 in). As the boundary spacing is increased, the affect of inner ring corner radius is reduced. The reason for this stress pattern becomes apparent when FEA model stress distribution contour plots are viewed. At small inner ring corner radii there are stress concentrations at the inner ring corners. These concentrations are reduced as the radius is increased until a radius is reached that creates stress concentrations at the sides of the inner ring. These new stress concentration areas increase non-uniform stress. This change in stress concentration areas is shown in Figures 5.16 and 5.17.

Table 5.5: Inner ring corner radius results

Case	D (mm)	Inner Radius (mm)	Von Mises Stress Values		
			Average (MPa)	Std Dev (MPa)	η (%)
IR-01	25.40	0	54.79	8.889	16.23
IR-02	25.40	5.08	53.85	7.297	13.55
IR-03	25.40	12.7	52.57	5.695	10.83
IR-04	25.40	25.4	50.05	4.068	8.13
IR-05	25.40	38.1	46.17	2.217	4.80
IR-06	25.40	43.18	44.36	1.973	4.45
IR-07	25.40	48.26	42.11	1.857	4.41
IR-08	25.40	50.8	40.97	2.128	5.19
IR-11	31.75	0	24.82	2.650	10.68
IR-12	31.75	5.08	24.49	2.132	8.70
IR-13	31.75	12.7	24.09	1.691	7.02
IR-14	31.75	25.4	23.17	1.180	5.09
IR-15	31.75	38.1	21.80	0.664	3.04
IR-16	31.75	43.18	21.14	0.549	2.60
IR-17	31.75	48.26	20.34	0.519	2.55
IR-18	31.75	50.8	19.94	0.585	2.93
IR-21	38.10	0	14.88	1.211	8.14
IR-22	38.10	5.08	14.72	0.968	6.58
IR-23	38.10	12.7	14.57	0.862	5.92
IR-24	38.10	25.4	14.08	0.561	3.98
IR-25	38.10	38.1	13.39	0.303	2.26
IR-26	38.10	43.18	13.06	0.243	1.86
IR-27	38.10	48.26	12.67	0.212	1.67
IR-28	38.10	50.8	12.47	0.238	1.91

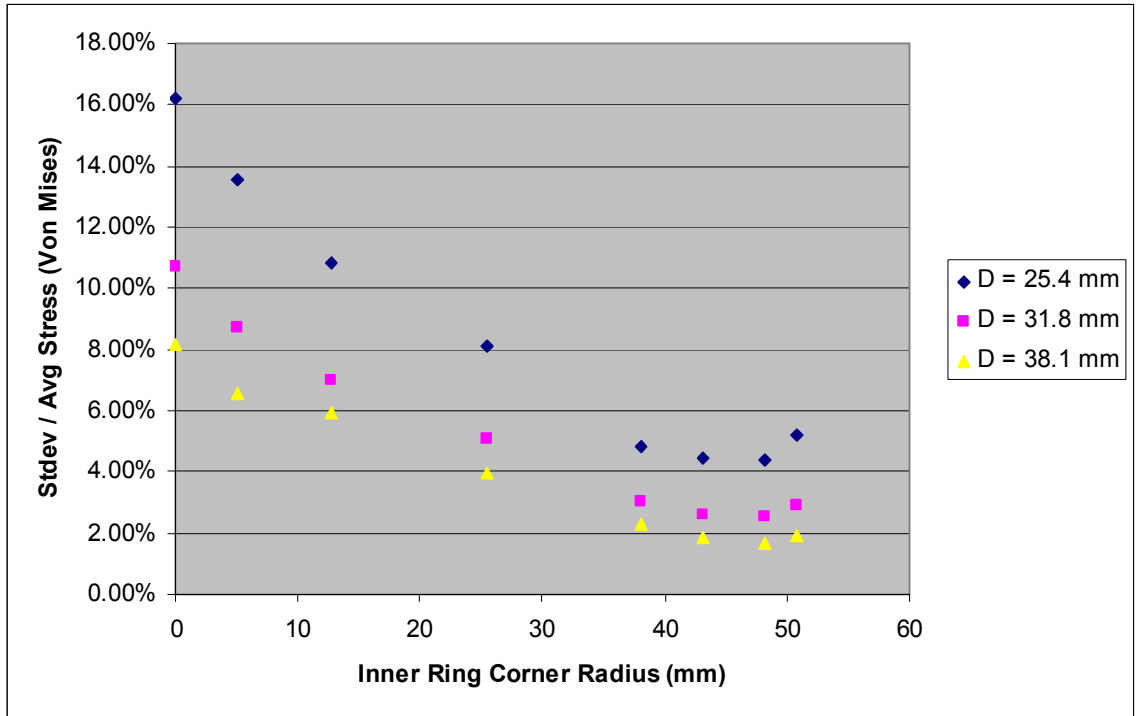


Figure 5.15: Plot of stress non-uniformity, η , versus inner boundary corner radius with respect to boundary spacing (D)

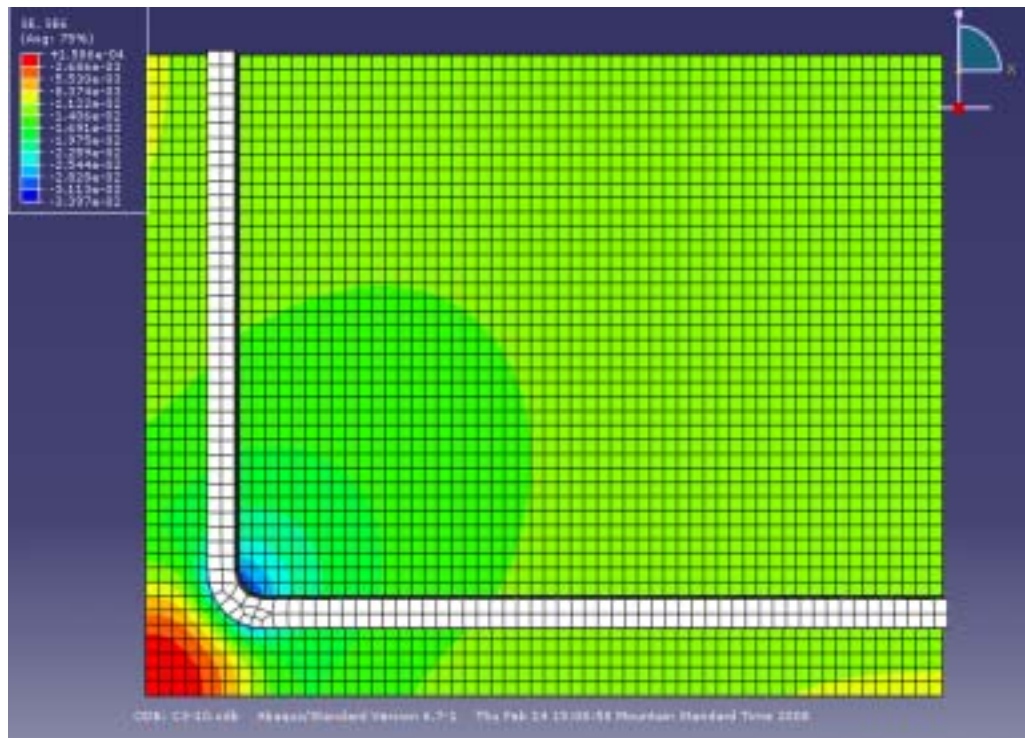


Figure 5.16: Small inner ring corner radius VM stress distribution (GPa) (notice the stress concentration at the inner ring corner)

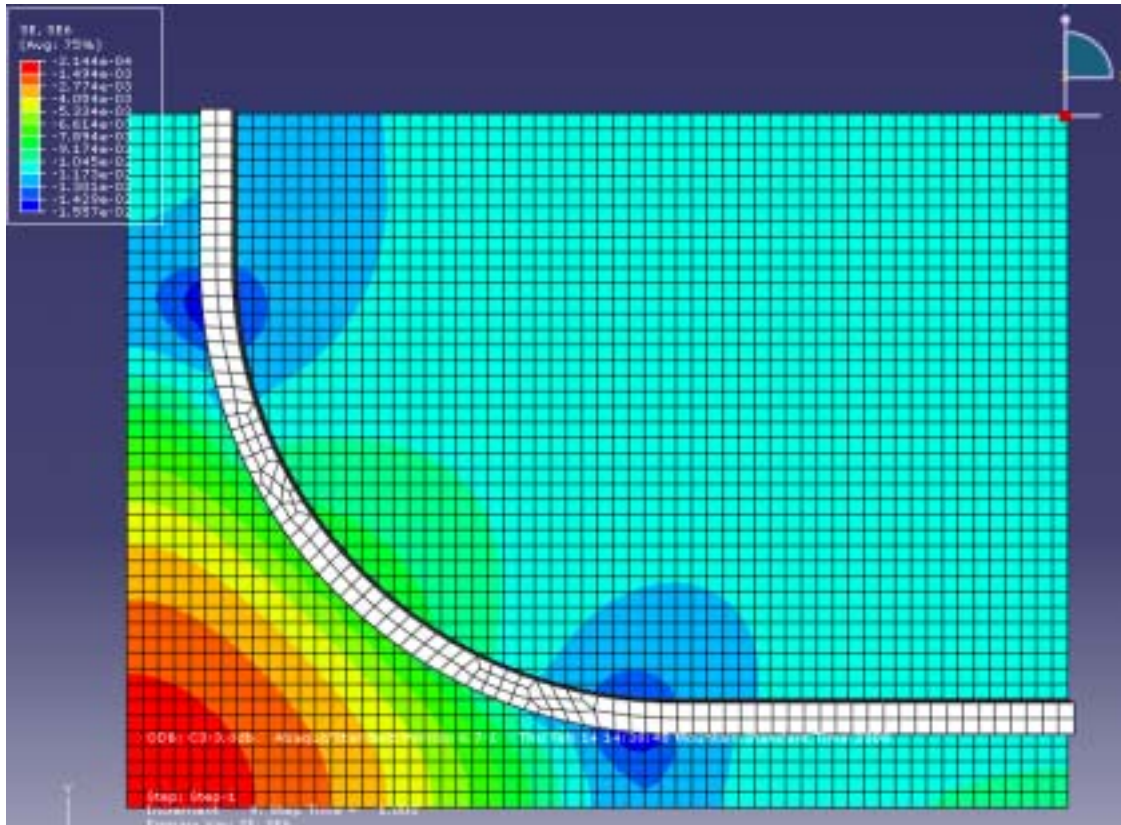


Figure 5.17: Large inner ring corner radius VM stress distribution (GPa)
(notice stress concentrations at sides of ring)

Inner to Outer Ring Boundary Spacing

Inner ring to outer boundary spacing model details are described in Chapter 4. Eighteen FEA models with three different size inner rings and varying inner to outer ring boundary spacing (D) were ran. The results of the models are shown in Table 5.6 and plotted in Figure 5.18. Cases BS-01, BS-02, BS-11, BS-12, BS-21, and BS-22 all had maximum Von Mises substrate stress values over 80 MPa leading to possible yielding.

Table 5.6: Inner ring to outer boundary spacing FEA results

		Von Mises Stress Values		
Case	D (mm)	Average (Mpa)	Std Dev (Mpa)	N (%)
BS-01	6.35	75.0321	8.8830	11.84
BS-02	8.89	49.9272	4.7961	9.61
BS-03	15.24	25.0080	1.7002	6.80
BS-04	21.59	15.5317	0.9337	6.01
BS-05	34.29	7.9355	0.4468	5.63
BS-06	46.99	4.9126	0.2804	5.71
BS-11	6.35	58.6509	8.5046	14.50
BS-12	8.89	39.5525	4.7401	11.98
BS-13	15.24	20.3289	1.6017	7.88
BS-14	21.59	12.8743	0.8146	6.33
BS-15	34.29	6.7593	0.4019	5.95
BS-16	46.99	4.2520	0.2467	5.80
BS-21	6.35	48.3879	8.5952	17.76
BS-22	8.89	32.8781	4.8728	14.82
BS-23	15.24	17.1671	1.7961	10.46
BS-24	21.59	11.0192	0.9121	8.28
BS-25	34.29	5.9039	0.4113	6.97
BS-26	46.99	3.7683	0.2276	6.04

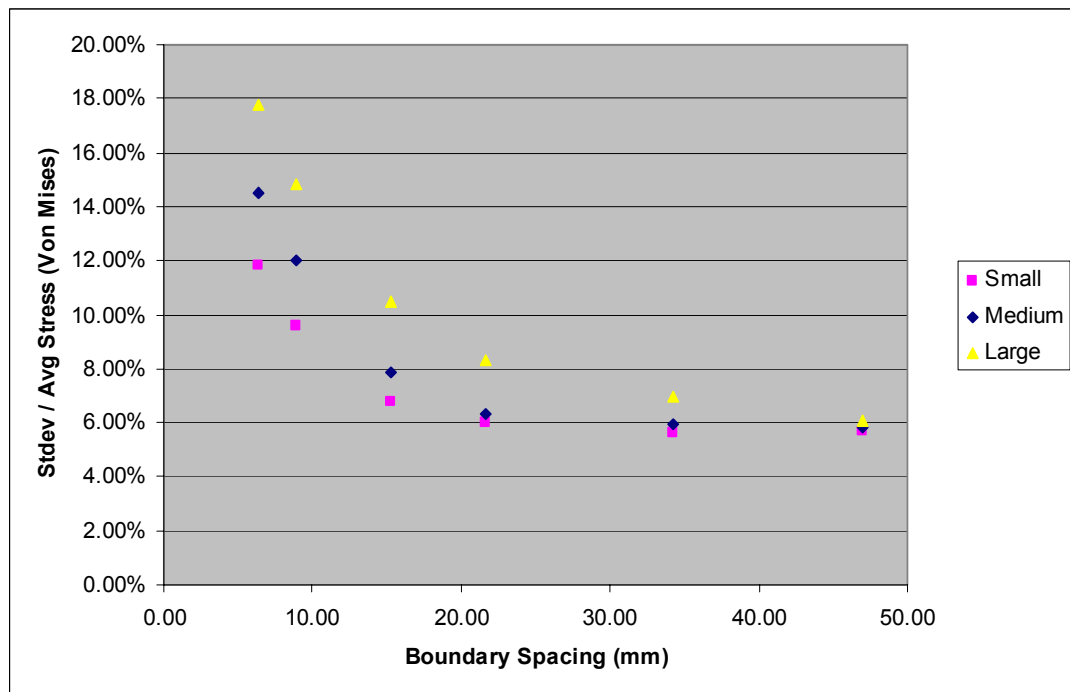
Figure 5.18: Plot of stress non-uniformity, η , versus inner to outer boundary spacing with respect to small, medium, and large inner ring dimensions

Figure 5.18 shows how boundary spacing has a significant effect on amount of non-uniform stress. The largest models show the most improvement with a three times decrease in uneven stress from case BS-21 to BS-26. The smallest models show a factor of two decrease from BS-01 to BS-06. Interestingly, all models seem to converge on an equal amount of stress non-uniformity as the boundary spacing is increased. The amount of average stress decreases at a much more rapid pace with an over 14 times decrease of average stress on all model sets. Increasing boundary spacing reduces the amount of non-uniform stress and significantly reduces average stress values that could lead to pellicle failure.

Catenary Outer Boundary

Catenary outer boundary model details are described in Chapter 4. The Von Mises stress and standard deviation of stress results for the five catenary outer boundary FEA models are shown in Table 5.7. The amount of curvature column in Table 5.7 is a qualitative rating from 1 (least amount of boundary curvature) to 5 (greatest amount of boundary curvature). Cases CA-01 and CA-02 had maximum Von Mises substrate stress values over 80 MPa leading to possible yielding.

Table 5.7: FEA catenary boundary models

Case	Amount of Curvature	Von Mises Stress Values		
		Average (MPa)	Std Dev (Mpa)	N (%)
CA-01	1	36.649	4.507	12.30
CA-02	2	32.731	3.469	10.60
CA-03	5	22.738	3.312	14.56
CA-04	4	31.561	2.798	8.87
CA-05	3	33.070	2.572	7.78

Table 5.7 shows how the catenary boundary can be effective at reducing non-uniform stress. The reduction of non-uniform stress is achieved by finding the exact parabolic catenary shape that provides a uniform stress around the inner ring. Too little curvature creates stress concentrations at the corners while too much curvature creates stress distributions at the centers of each side on the inner ring. FEA models of too little, suitable, and excessive catenary curvature are shown in Figures 5.19 through 5.21 (only the effective aperture is shown and has jagged edges due to meshing the catenary boundaries that are not shown.)

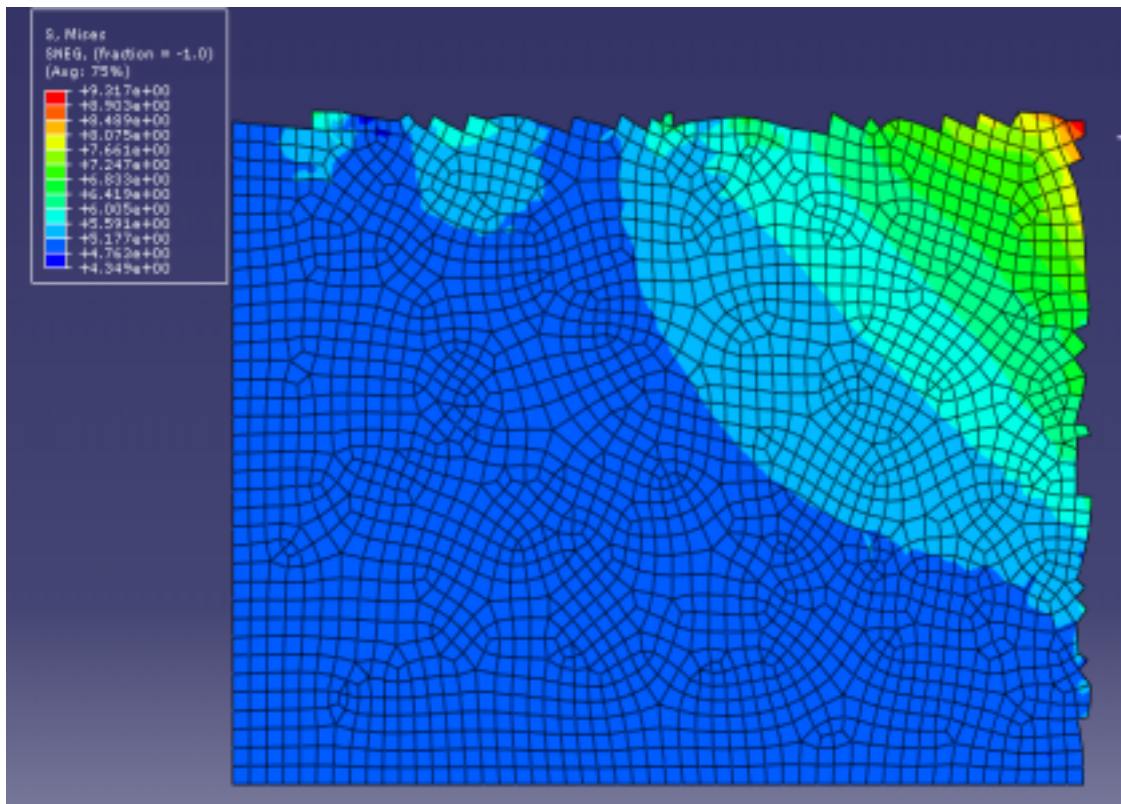


Figure 5.19: Too little catenary boundary curvature model (Case CA-01)
(Notice stress concentration at corner)

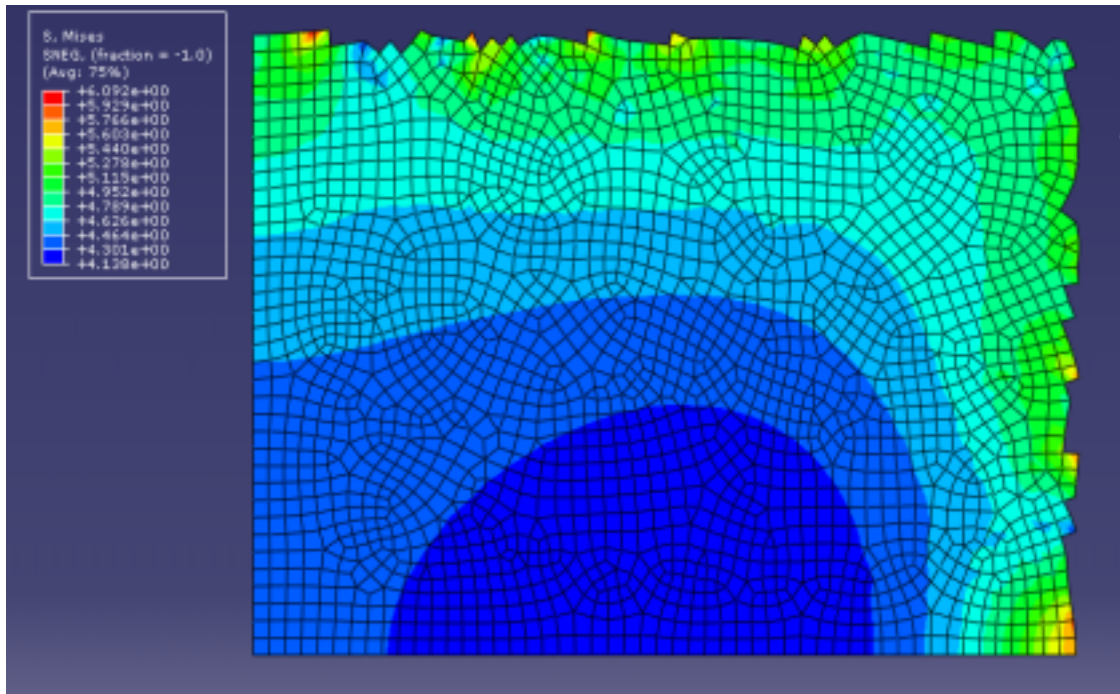


Figure 5.20: Suitable catenary boundary curvature model (Case CA-05)
(Notice approximate even stress along boundary)

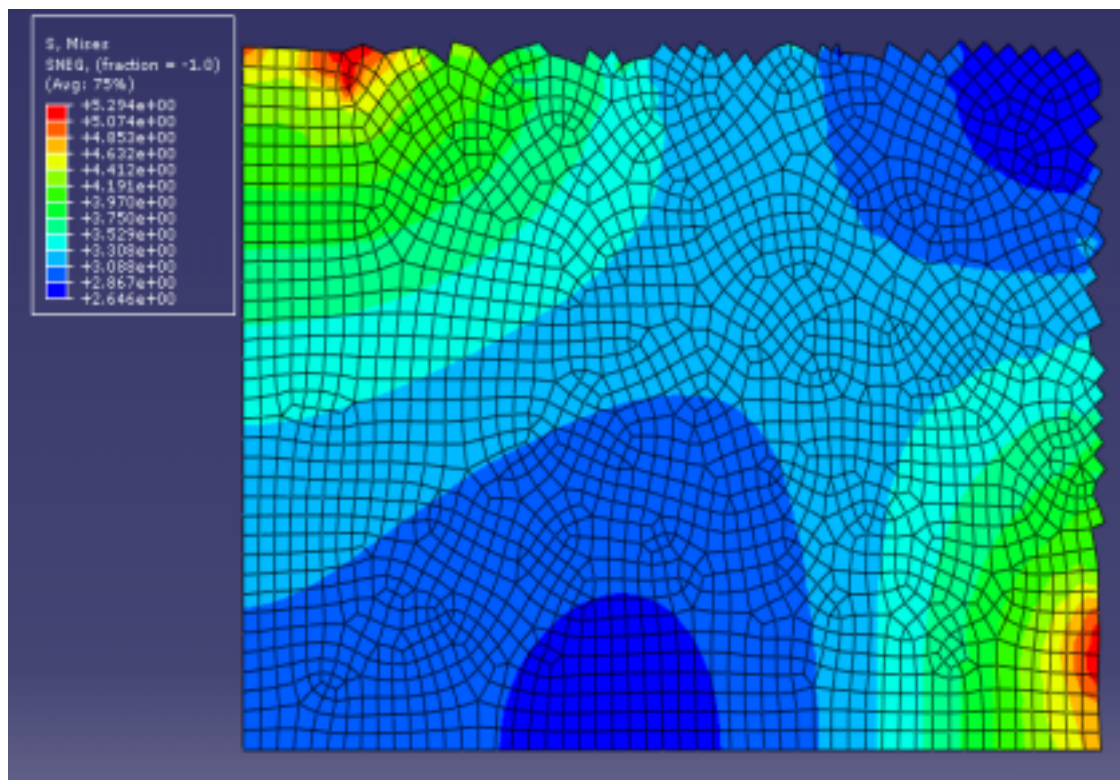


Figure 5.21: Excessive catenary boundary curvature model (Case CA-03)
(Notice stress concentrations at sides and corner)

Ideal Fixture Geometry

The ideal fixture geometry will vary depending on many parameters, including; allowable outer boundary size, required clear aperture, manufacturability, mount requirements, and more. Important parameters studied that could improve optical quality by fixture geometry were inner ring corner radii, boundary spacing, and catenary outer boundary. Geometry factors that were insignificant or degraded optical quality were pellicle thickness, inner ring displacement, and outer boundary corner radii. It is possible to combine elements to further increase optical quality. Figure 5.22 shows a combination of the best performing catenary boundary with the best performing inner ring radius. The resulting value of η was 4.04%, which is much lower than either case individually. Increasing the boundary spacing would further improve surface quality. The ideal rectangular fixture geometry would have large inner ring radii with large boundary spacing and possibly a catenary outer boundary.

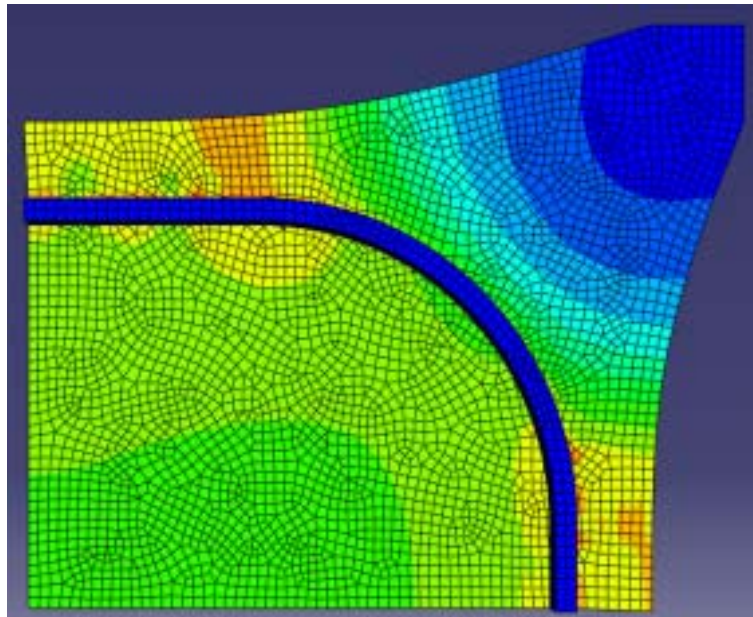


Figure 5.22: Catenary boundary with inner ring corner radius FEA model

CONCLUSION / RECOMENDATIONS

Conclusion

Undesirable optical quality in the rectangular pellicle beam splitter is caused in part or in whole by the rectangular inner and outer boundary shape. The rectangular shape causes high stress concentrations in the corners leading to a non-uniform stress over the pellicle. This non-uniform stress causes non-uniform pellicle thickness. Pellicles, as transmissive optics, are sensitive to thickness variations and having a non-uniform thickness degrades optical quality.

Finite element analysis has shown that inner ring corner radii and inner ring to outer boundary spacing are the two most effective geometry factors for reducing stress non-uniformity. Using a catenary outer boundary has also proven effective. Other geometry factors have minimal or negative impact on optical surface quality. Combining geometry factors can further decrease stress non-uniformity. The ideal fixture geometry will vary for each case, but in general, having a large radii inner ring and large boundary spacing will improve the optical quality of a rectangular beam splitter.

The initial prototypes had undesirable optical quality evidenced by fringe patterns radiating from the corners. Another prototype was made with a compliant boundary but it did not effectively improve optical quality. Finite element analysis showed using a compliant boundary would not likely improve these fringe patterns. Modeling showed that having large inner ring radii is very effective at improving optical quality. A

prototype with large inner ring radii was made and found to have significantly improved optical quality with the fringe patterns radiating from the corners virtually eliminated.

Recommendations

The finite element analysis models used were based on many assumptions in order to simplify the models. These assumptions were used to save time, but more accurate models could be made. The models could incorporate details such as a fully modeled outer boundary, fixture surface quality, inner ring displacement by screws, pellicle thickness variations from processing, and bond details. Including friction between the pellicle and inner ring could have a significant affect on the results. The data created could be normalized to develop design charts. More models could be made to view the affects of having different horizontal and vertical boundary spacing's. Finally, a more in-depth correlation between the FEA models and prototypes should be completed.

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APPENDICES

APPENDIX A

ABBREVIATED ABAQUS FEA INPUT FILE

Input file notes are shown in bold

```

*Heading
** Job name: C1-10 Model name: Model-1
*Preprint, echo=NO, model=NO, history=NO, contact=NO
**
** PARTS
**
*Part, name=ring
Define inner ring node and element locations (ABAQUS CAE created)
*Node
    1, 2.29999995, 0.449999988, 0.
    2, 2.29999995, 0.349999994, 0.
    3, 0.349999994, 0.349999994, 0.
    .
    .
    594, 0.399959445, 1.700000076, 0.100000001
*Element, type=C3D8R Inner ring element type
    1, 2, 7, 135, 8, 200, 205, 333, 206
    2, 1, 99, 135, 7, 199, 297, 333, 205
    .
    .
    260, 396, 270, 269, 394, 594, 468, 467, 592
*Nset, nset=_PickedSet2, internal, generate
    1, 594, 1
*Elset, elset=_PickedSet2, internal, generate
    1, 260, 1
** Section: ring
Define inner ring material (aluminum)
*Solid Section, elset=_PickedSet2, material=Al
1.,
*End Part
**
Define membrane node and element locations (ABAQUS CAE created)
*Part, name=shell
*Node
    1, 3., -0.5, 0.
    2, 3., -0.449999988, 0.
    3, 3., -0.400000006, 0.
    .
    .
    3752, -0.300000012, 2.25, 0.
*Element, type=S4R Membrane element type
    1, 1, 2, 58, 57
    2, 2, 3, 59, 58
    3, 3, 4, 60, 59
    .
    .
    3630, 3695, 3696, 3752, 3751
*Nset, nset=_PickedSet2, internal, generate
    1, 3752, 1
*Elset, elset=_PickedSet2, internal, generate
    1, 3630, 1
** Section: shell
Define membrane material (NPS) and thickness (0.1 in)
*Shell Section, elset=_PickedSet2, material=NPS

```

0.001, 5

*End Part

**

**

Assemble inner ring and membrane

** ASSEMBLY

**

*Assembly, name=Assembly

**

*Instance, name=shell-1, part=shell

*End Instance

**

*Instance, name=ring-1, part=ring
0.7, 0.5, 0.

*End Instance

**

Define element and node sets for applying boundary conditions and loads

*Nset, nset=_PickedSet10, internal, instance=shell-1, generate

56, 3752, 56

*Elset, elset=_PickedSet10, internal, instance=shell-1, generate

55, 3630, 55

*Nset, nset=_PickedSet11, internal, instance=ring-1, generate

397, 594, 1

*Elset, elset=_PickedSet11, internal, instance=ring-1, generate

131, 260, 1

*Nset, nset=_PickedSet12, internal, instance=ring-1, generate

397, 594, 1

*Elset, elset=_PickedSet12, internal, instance=ring-1, generate

131, 260, 1

*Nset, nset=_PickedSet13, internal, instance=shell-1

1, 57, ..., 3751, 3752

*Elset, elset=_PickedSet13, internal, instance=shell-1

1, 56, ..., 3629, 3630

*Nset, nset=_PickedSet14, internal, instance=shell-1, generate

1, 56, 1

*Elset, elset=_PickedSet14, internal, instance=shell-1, generate

1, 55, 1

*Elset, elset=__PickedSurf6_SPOS, internal, instance=shell-1, generate

1, 3630, 1

*Surface, type=ELEMENT, name=_PickedSurf6, internal

__PickedSurf6_SPOS, SPOS

*Elset, elset=__PickedSurf7_S1, internal, instance=ring-1, generate

1, 130, 1

*Surface, type=ELEMENT, name=_PickedSurf7, internal

__PickedSurf7_S1, S1

*End Assembly

**

Define materials and their properties

** MATERIALS

**

*Material, name=Al

*Elastic

10000., 0.33

*Material, name=NPS

```

*Elastic
663., 0.4
**

Define interaction between inner ring and membrane as frictionless
** INTERACTION PROPERTIES
**
*Surface Interaction, name=IntProp-1
1.,
*Friction
0.,
**
** BOUNDARY CONDITIONS
**

Pin outer boundary
** Name: BC-1 Type: Symmetry/Antisymmetry/Encastre
*Boundary
_PickedSet13, PINNED

Define x-axis of symmetry
** Name: BC-2 Type: Symmetry/Antisymmetry/Encastre
*Boundary
_PickedSet14, XSYMM

Define y-axis of symmetry
** Name: BC-3 Type: Symmetry/Antisymmetry/Encastre
*Boundary
_PickedSet10, YSYMM

Constrain inner ring to only z-direction movement
** Name: BC-4 Type: Symmetry/Antisymmetry/Encastre
*Boundary
_PickedSet11, ZASYMM
**

Define contact between inner ring and membrane
** INTERACTIONS
**
** Interaction: Int-1
*Contact Pair, interaction=IntProp-1, small sliding, type=SURFACE TO
SURFACE, no thickness
_PickedSurf7, _PickedSurf6
** -----
**

Create step to apply inner ring displacement
** STEP: Step-1
**
*Step, name=Step-1, nlgeom=YES Turn on non-linear geometry solvers
*Static
1., 1., 1e-05, 1.
**

Displace inner ring against membrane in z-direction (-0.1 in)
** BOUNDARY CONDITIONS
**
** Name: BC-5 Type: Displacement/Rotation
*Boundary
_PickedSet12, 3, 3, -0.1

```

**

Define desired output results (all available stress and strain)

** OUTPUT REQUESTS

**

*Restart, write, frequency=0

**

** FIELD OUTPUT: F-Output-1

**

*Output, field, frequency=99999

*Element Output, directions=YES

ALPHA, CTSHR, DAMAGEC, DAMAGET, E, EE, ER, IE, LE, MISESMAX, NE, PE,

PEEQ, PEEQMAX, PEEQT, PEMAG

PEQC, PS, S, SALPHA, SDEG, SE, SEE, SEP, SEPE, SPE, THE, TSHR, VE,

VEEQ, VS

**

** HISTORY OUTPUT: H-Output-1

**

*Output, history, variable=PRESELECT, frequency=99999

*End Step

APPENDIX B

ABAQUS FEA COMPLIANT BOUNDARY SPRING COMAND

Input file notes are shown in bold

```
*Heading
** Job name: spring2_try Model name: Model-1
*Preprint, echo=NO, model=NO, history=NO, contact=NO
**
```

Define nodes to apply springs to

```
*Element, type=Spring1, elset=Springs/Dashpots-1-spring
1, shell-1.2
2, shell-1.3
3, shell-1.4
. . .
140, shell-1.1271
```

Define spring constant in x-direction (1)

```
*Spring, elset=Springs/Dashpots-1-spring
1
0.076
```

Define nodes to apply springs to (Same as above)

```
*Element, type=Spring1, elset=Springs/Dashpots-2-spring
141, shell-1.2
142, shell-1.3
143, shell-1.4
. . .
280, shell-1.1271
```

Define spring constant in y-direction (2)

```
*Spring, elset=Springs/Dashpots-2-spring
2
0.076
*End Assembly
```

Z-direction stiffness was considered infinitely stiff

APPENDIX C

MEMBRANE IN-PLANE STRESS TO THROUGH THICKNESS
STRAIN RELATIONSHIP

It is important to relate in-plane stress non-uniformity with membrane through thickness non-uniformity. Two sets of models were used to compare in-plane stress to through thickness strain. Inner ring displacement (section 4.4) and boundary spacing (Section 4.8) models from previous analysis were used. Then through thickness strain (z-direction) was output from the same results aperture as in-plane stress. The average value of the strain was divided by the standard deviation of the strain. This gave a value to compare to the in-plane average VM stress divided by the standard deviation of VM stress (see Section 5.1). Tables D.1 and D.2 show results of in-plane stress and through thickness strain. These values are plotted in Figures D.1 and D.2

Case	Ring Displacement (mm)	In-Plane Stress	Through Thickness Strain
		Stdev/Avg (%)	Stdev/Avg (%)
RD-01	1.272	7.01	6.14
RD-02	2.544	7.17	6.21
RD-03	3.816	7.49	6.34
RD-04	5.088	7.54	6.67

Table D.1: Comparison of stress and strain for inner ring displacement models

Case	D (mm)	In-Plane Stress	Through Thickness Strain
		Stdev/Avg (%)	Stdev/Avg (%)
BS-11	6.35	14.50	13.57
BS-12	8.89	11.98	11.17
BS-13	15.24	7.88	8.10
BS-14	21.59	6.33	6.67
BS-15	34.29	5.95	6.10
BS-16	46.99	5.80	5.58

Table D.2: Comparison of stress and strain for boundary spacing models

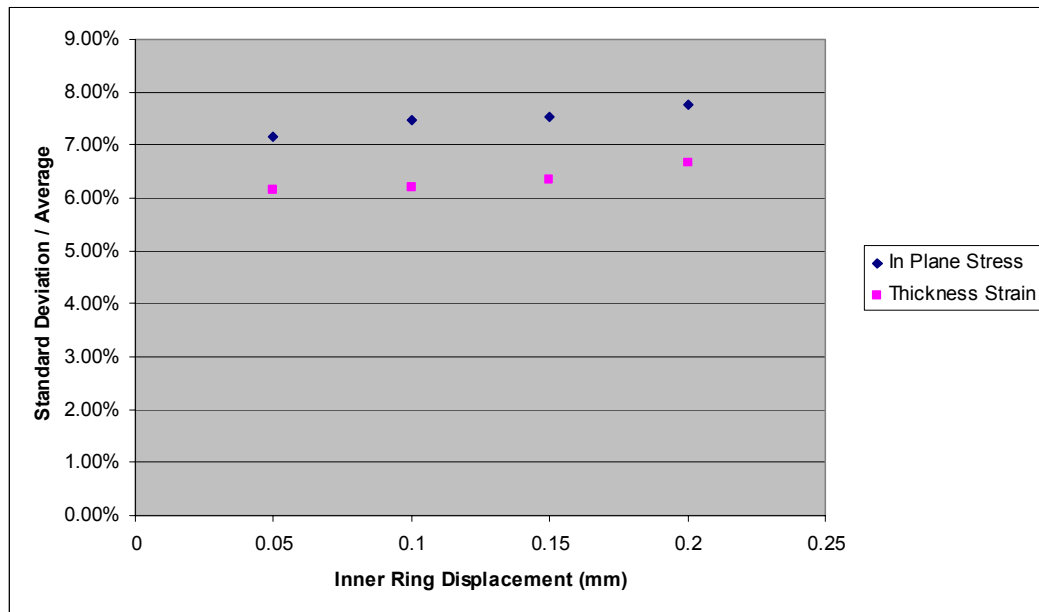


Figure D.1: Plot of stress and strain for inner ring displacement models

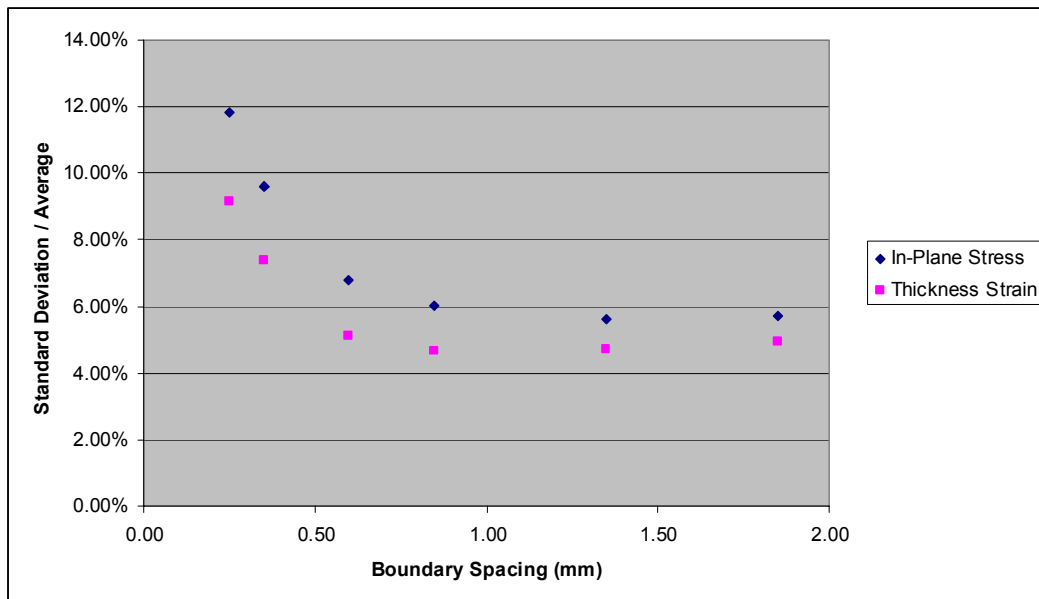


Figure D.2: Plot of stress and strain for boundary spacing models

Figures D.1 and D.2 show how through thickness strain follows closely the trend of in-plane stress. This shows that in-plane stress non-uniformity is directly related to through thickness strain non-uniformity.

APPENDIX D

FEA MODEL MESH DENSITY STUDY

All FEA models used the same mesh density leading to more elements in larger models than smaller one. In some cases with large boundary spacing there were three times as many elements as smaller models. To study the effect of the mesh density on results the boundary spacing models were used (see Section 4.8). The largest models BS-21 to BS-26 were re-created with mesh count similar to that of the smallest models BS-01 to BS-06. This reduced the mesh density by roughly a factor of three. Figure E.1 shows the results of the original models BS-21 to BS-26 plotted against those with a lower mesh density. All results were within 5% of each other confirming a safe assumption to use constant mesh density instead of a constant number of elements.

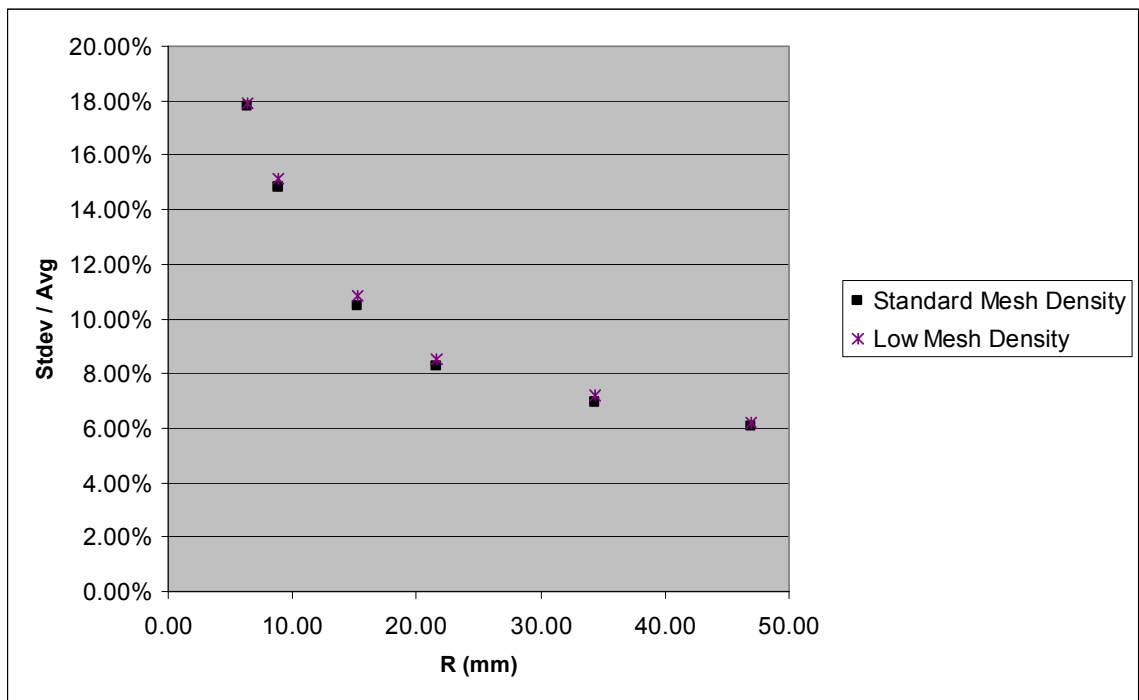


Figure E.1: Plot of standard and low mesh densities affect on results