Chapter 5

Instrumentation Case Study: Thermal Performance of a Modular Structure for a High-Precision Microscope

This chapter describes a novel segmented design for an instrumentation structure based on modules connected by canoe ball kinematic couplings. The design is applied to a high-precision microscope for single-molecule experiments, under primary development at the University of Illinois. The modular segmented design is a series of metallic rings, and is shown to be significantly less sensitive to deformation from uneven thermal disturbances than a similar structure consisting of a single full-length ring, or a traditional non-symmetric microscope structure. Sub-micron repeatability of the kinematic coupling interfaces enables the structure to be disassembled and reassembled without re-calibration, and pre-calibrated ring modules with different optical components can be installed for quick changeover between experiments. The design and thermal experiment results are presented here, with the repeatability assessments as ongoing work to be presented in a future publication.

5.1 Overview of the High Precision Microscope Project

Recent developments in microscopy have focused on using microscopes for quantitative measurements in addition to imaging. Spectroscopy has also become an important tool in microscopy, requiring the use of additional optical devices in conjunction with the microscope itself. Furthermore, technological advances have made single molecule detection possible, driving interest in measurements with sub-nanometer precision [1].

These new applications require performance beyond the capabilities of conventional microscopes, and have revealed the need for improvement in the optomechanics of microscopes, in three main areas:

1. Resolution - The ability to adjust the optics and sample on a nanometer scale

2. Flexibility - The ability to use a wide assortment of peripheral optics and devices; the ability to use several different experimental modes simultaneously; and the ability to rapidly reconfigure the microscope without the need for re-alignment.

3. Stability - Resistance to vibration, acoustic, and thermal noise; hence the ability to maintain an...
optical alignment within nanometers over the course of hours [1].

The High Precision Microscope (HPM) project, headquartered at the University of Illinois Laboratory for Fluorescence Dynamics (LFD) and funded by the National Institutes of Health (NIH), is developing a novel system and a set of optics modules to address each of these shortcomings through the construction of a new microscope for high resolution, single molecule experiments. The microscope will be equipped with the experimental capabilities of being a symmetric upright and inverted microscope, and support advanced optical capabilities such as wide field Differential Interference Contrast (DIC) imaging, wide field fluorescence imaging, scanning fluorescence imaging, Laser DIC microinterferometry, quadrant position detection, and Total Internal Reflection (TIR) illumination [1]. These capabilities are integrated together such
that many of them may be used simultaneously. Ideally, optical modules will be interchangeable with sufficient accuracy to require no calibration upon reconfiguration of the setup. A concept model of the HPM is shown in Figure 5.1.

When this microscope is used for single molecule experiments, the most important specification is the position stability of a spot on the object plane, formed by the focus of the laser. UIUC LFD is collaborating with the MIT Precision Engineering Research Group (PERG) in mechanical design and packaging of the microscope. The MIT PERG is working to design and test a modular, kinematically coupled segmented structure, verifying its improved thermal stability and mechanical repeatability over both traditional and single-piece tubular microscope structure designs.

This chapter presents the segmented structural design for the HPM, and methodology and results for assessments of its dimensional stability under asymmetric thermal disturbances. Its goal of mechanical repeatability and exchangeability on disassembly, reassembly and reconfiguration is well grounded in past assessments of canoe ball kinematic coupling performance, and detailed assessments of the performance of the serial kinematic chain of the microscope are underway and will be presented at a later date.

5.2 Design and Theoretical Basis of the Modular Structure

The HPM structure is the assembly that will hold all of the optical elements in place, thus determining their position and fit, and the relative alignment pass-through of optical signals. In this respect, the primary requirements of the structure are to:

1. Hold the optical and mechanical elements precisely together and minimize their sensitivity to thermal drift, mechanical vibration, and acoustic noise.

2. Support several optical paths for the peripheral optics and devices.

3. Allow easy access to the components so they may be aligned, adjusted, and cleaned without removal.

4. Be air and light tight, although being vacuum tight is not required.

5. Be arranged in such a way as to reduce stray light within the system.

To meet these requirements, a modular tubular structure was proposed, with canoe ball kinematic couplings connecting the structural segments. The three main projected advantages of this design are:
1. Stability with respect to thermal creep because the tube structure distributes heat so that the tube expands with significantly less circumferential thermal gradient than a one-piece tubular structure.

2. Precision alignment of the optical axes, enforced by the inherent repeatability and geometric error averaging behavior of the kinematic couplings between the segments.

3. Easy reconfiguration to accommodate different optical assemblies.

Under normal laboratory conditions, this design seeks to meet performance goals of:

1. Less than 5 nm deviation of the optical beam at the sample stage from angular drift of the stack structure, caused by gradual room temperature fluctuations of less than 0.5 °C.

2. Relative radial repeatability of the optical axes of 0.2 microns when the stack is disassembled and reassembled in an identical serial configuration. If interchangeable modules are calibrated with position and orientation errors of the ball and groove placements, 0.2 micron exchangeability shall be achievable when these error offsets are considered.

The design breaks the structure into vertical sections, a series of aluminum tube rings connected by highly repeatable canoe ball kinematic couplings, as shown in Figure 5.2. By segmenting the structure, it is hypothesized that an incident asymmetric thermal disturbance will be directed circumferentially around the structure, and significant axial heat flow will be prevented by the air gaps between the tubes and the minimal coupling point contacts between the tubes. Hence, by maximizing the angular uniformity of the temperature distribution of the structure, the asymmetry of axial thermal expansion of the structure will be minimized. In a microscope, non-uniform axial thermal expansion causes in-plane misalignment of the optical beam axes at the point of examination; even drift of a few nanometers can cause loss of the image in single molecule experiments. In this respect, the infinity corrective objectives used in present microscopes can deal with uniform axial expansion, and radial expansion is not an issue as long as the center point where the objectives reside remains in the same radial position as the stage below it. Here, an equal angle three-groove arrangement of kinematic couplings would provide uniform radial expansion and geometrically-averaged motion of the center point by constraining the ball sets to slide in the grooves when the tubes expand.

5.2.1 Specifications of Prototype Structures

The prototype segmented stack structure for the thermal stability and mechanical repeatability experiments consists of five hollow cylindrical 6061-T651 Aluminum tubes, 12” outside diameter and 9” inside
diameter, with equilaterally triangular hole and spot-face patterns on each end to accommodate the kinematic couplings. The center tube is 2.56” long, while the remaining four tubes are identical in feature geometry, but 4.24” long. These dimensions were scaled down (to decrease manufacturing cost) from the original dimension set with 16” outside diameter, proposed by the UIUC LFD for the operational microscope; however, critical dimensionless ratios relating to heat flow through the geometry were preserved. The geometrically similar control structure, a single tube with the same inner diameter, outer diameter, and total length as the coupled series of segments, was also procured.

Figure 5.2: Exploded model of segmented structure including two canoe ball sets.

Figure 5.3: Close view of canoe ball interfaces between segments.
The canoe ball couplings, used as interconnects between the segments of the former structure, and between the ends of the tube regions and the mounting plates of both structures, were machined from AISI 420 stainless steel, hardened to Rockwell C50-55 and CNC precision ground to fine surface roughness. The male ball units have a spherical surface radius of 0.25 m, and the female groove units are standard 45-degree flat vees. The couplings were aligned to the seat mounts using 3.0 mm diameter spring pins, and lightly (0.02 mm $\Delta d$) press-fit into place. Contrary to the robot interface application, stress-limiting design of the couplings was not a factor; sizes were chosen to take advantage of high repeatability with large surface radius and fine surface finish, and for stiffness to maintain high natural frequencies.

For testing, both tube assemblies were placed between the same set of identical top and base plates, and tightened in compression with full-length threaded rods. To distribute the compressive load equally among the couplings, an additional plate was added at the top of the structure, to which the threaded rods were bolted through stacks of belleville spring washers, and the load was transmitted to the stack by a centrally seated 1” diameter steel tooling ball.

5.2.2 Supporting Heat Transfer Theory

Having presented the concept of a segmented, kinematically coupled design for instrumentation structures, its performance characteristics can now be explained in terms of general heat transfer and mechanics relations. First, the theorized ability of the segments to enforce greater circumferential uniformity of temperature than the single-piece structure is seen by examining the constant temperature profiles when an ideal point-located disturbance is applied to one side. As shown in Figure 5.4, the temperature profiles on the single-piece tube (considering length far greater than diameter) are nearly circular in side view, and flow the disturbance equally in axial and circumferential directions. On the other hand, the far lesser length to diameter ratio of the short segment constrains the axial heat flow, forcing the contours to show constant temperature bands in the circumferential direction. Hence, the difference in average temperature between the heated and non-heated sides of the long tube is greater than that on the short tube, and the greater total variance in temperature between the distant sides of the single-piece structure creates a greater discrepancy in the non-uniformity of thermal expansion.
The one-dimensional thermal expansion of a body with length $L_o$, subject to a uniform temperature increase $\Delta T$, is:

$$\frac{\Delta L}{L} = \varepsilon_{thermal} = \alpha_t \Delta T,$$

(5.1)

where $\alpha_t$ is the material coefficient of thermal expansion. Relating this to the difference in length $\delta$ between opposite sides of a cylinder when a thermal disturbance is applied to one side:

$$\delta = \alpha_t L_o (\bar{T}_h - \bar{T}_n) = \alpha_t \left[ \int_0^{L_o} (T_h(z) - T_n(z)) dz \right],$$

(5.2)

where the temperatures on the heated ($T_h$) and non-heated ($T_n$) sides are expressed as averages or definite integrals.
The gross error motion that causes misalignment of the optical axes is tilt of the top of the structure relative to a fixed bottom plane. Geometrically, this angle relates to $\delta$ by:

$$\theta_{tilt} = \tan \left( \frac{\delta}{D_s} \right), \quad (5.3)$$

where $D_s$ is the diameter of the structure at the supports of the top plate. $\theta_{tilt}$ equals the angle of misalignment of the optical axes; hence, $\delta_{obj}$, the translational error of the objectives at the sample stage is:

$$\delta_{obj} = L_s \frac{\delta}{D_s}, \quad (5.4)$$

where $L_s$ is the axial length from the top plate to the sample position. Combining relations (5.2) and (5.4), the translational error at the sample position relates to the mean temperature difference through:

$$\delta_{obj} = L_s \frac{\alpha_{L_o} (T_h - T_a)}{D_s}. \quad (5.5)$$

Substituting values for the prototype design, the mean temperature difference causing 5 nm drift at the objectives is $0.00047 \degree C$. Although this is an extremely small difference, one which would be extremely difficult to control for experimentation, the primary goal of this study is to show the magnitude of benefit of the segmented design over the one-piece design; this is not complicated by applying an artificially large thermal disturbance since the error motion scales linearly with the temperature difference across the structure.

Next, the steady-state temperature difference is related to the magnitude of the thermal disturbance, $Q$, by the generalized thermal resistance, $R$, through:

$$Q = \frac{(T_h - T_a)}{R}. \quad (5.6)$$

Treating the circumferential path of heat flow as one through a generalized linear body with effective area $A$, effective length $L$, and material thermal conductivity $k$,

$$Q = \frac{(T_h - T_a)}{L} Ak. \quad (5.7)$$

Rearranging (5.7) and substituting into (5.5):
Therefore, with the goal to minimize $\delta_{obj}$ for a chosen structure geometry, the best material for steady-state performance is one with maximum thermal conductivity per unit of tendency to thermally expand, $k/\alpha_t$. Higher $k$ decreases the steady-state temperature difference between sides of the structure, while higher $\alpha_t$ represents a greater magnification into translational error at the sample.

However, since in a normal laboratory the structure will conceivably never reach a steady-state, constantly being subject to small thermal disturbances convected by air currents, the transient performance of the design is of greater importance. While it is cumbersome to express the transient temperature profile for the annular geometry of this structure in closed form, it is known that the penetration depth of temperature change through the structure $t$ seconds after the thermal disturbance begins is related to the Fourier number, Fo:

$$Fo = \frac{\rho c_p}{\alpha L^2},$$

where $\alpha$ is the material thermal diffusivity and $L$ is a generalized characteristic length. Assuming fixed geometry, the optimal structure material for transient performance is one with maximum $\alpha/\alpha_t$. Values of the steady-state and transient performance indices, $k/\alpha_t$ and $\alpha/\alpha_t$, are given for candidate metals in Table 5.1.

<table>
<thead>
<tr>
<th>Material</th>
<th>Steady-state ($k/\alpha_t$, W/m-K)</th>
<th>Transient ($\alpha/\alpha_t$, m$^2$/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum (6061-T651)</td>
<td>7.0 x 10$^6$</td>
<td>2.8</td>
</tr>
<tr>
<td>Copper</td>
<td>2.4 x 10$^7$</td>
<td>6.9</td>
</tr>
<tr>
<td>Brass</td>
<td>5.8 x 10$^6$</td>
<td>1.8</td>
</tr>
<tr>
<td>Stainless Steel (AISI 410)</td>
<td>1.9 x 10$^6$</td>
<td>0.54</td>
</tr>
</tbody>
</table>

Table 5.1: Steady-state and transient performance index values for various metals.

Among the materials surveyed, copper is conclusively the best choice, with over twice the transient performance index value of aluminum. Although for material and manufacturing cost reasons, aluminum was chosen for the prototype setup, superior performance would result if copper is chosen for the final HPM. This is confirmed by simulation results presented later.
To justify choice of boundary conditions for finite element simulations to be discussed later, free convection approximations from the structure are now presented. When the structure sits in a relatively calm ambient environment, with surface temperature greater than the surrounding air, the dominant mode of heat loss is through thermobuoyant free convection. Since the center of the structure is closed by the top and base plates, it is assumed to behave like a solid cylinder with uniform surface temperature (ignoring local gradients due to application of the disturbance), for which the following closed-form relations are well known:

\[
Q_{ku} = \frac{A \kappa f (\text{Nu}_L) (T_s - T_\infty)}{L} \tag{5.10}
\]

\[
(\text{Nu}_L) = [(\text{Nu}_{L,i})^6 + (\text{Nu}_{L,i})^{\frac{1}{6}}]^6 \tag{5.11}
\]

\[
(\text{Nu}_{L,i}) = \frac{2.8}{\ln \left(1 + \frac{2.8}{a_1 (Ra_L)^{\frac{1}{3}}}\right)} \tag{5.12}
\]

\[
(\text{Nu}_{L,i}) = \frac{0.13 Pr^{0.22}}{1 + 0.61 Pr^{0.81}} (Ra_L)^{\frac{1}{3}} \tag{5.13}
\]

\[
a_1 = \frac{4}{3} \left[ \frac{0.503}{1 + \left(0.492 \frac{Pr}{Pr_i}\right)^{\frac{9}{16}}} \right]^{\frac{4}{9}} \tag{5.14}
\]

\[
(Ra_L) = \frac{g \beta_f (T_s - T_\infty) L^3}{\alpha_f \nu_f} \tag{5.15}
\]

\[
h = \frac{Q_{ku}}{A(T_s - T_\infty)} \tag{5.16}
\]

Here, \( g \) is the gravitational constant, \( \beta_f \) is the fluid coefficient of thermal expansion (reciprocal of absolute temperature for ideal gases), \( \alpha_f \) is the fluid thermal diffusivity, \( \nu_f \) is the fluid viscosity, and \( Pr \) is the fluid Prandtl number (0.69 for air at 300 K). Iteratively specifying a value for \( T_s \) using a spreadsheet, the steady-state convection coefficient \( h \) can be found for which the total surface convection loss \( Q_{ku} \) equals the magnitude of the thermal disturbance. For example, if 3W is applied, \( T_s = 296.8 \text{ K} \) (for \( T_{inf} = 295 \text{ K} \)) and \( h \) = 1.93 mK/W.
All heat transfer relations are referenced from Kaviany [2].

5.3 Thermal Stability Evaluation

The thermal stability of the HPM design was evaluated by applying circumferentially non-uniform thermal disturbances to the structures, and measuring the resulting circumferential and axial temperature distributions, and angular distortion of the top mounting plate. While the enforced disturbances were far stronger than those customary for a normal laboratory environment for a microscope, the results give an ordinal performance comparison between the designs, and validate a finite element model used for subsequent experiments and design optimization.

5.3.1 Experimentation Setup and Procedure

5.3.1.1 Temperature Metrology

The temperature distribution on the structure was monitored using fifty-two three-wire platinum RTDs (resistive temperature detectors), procured pre-assembled from National Instruments. The RTD’s were wired to a 16-bit PCMCIA computer data acquisition card through a NI SCXI-1000 chassis containing four SCXI-1122 multiplexer modules with SCXI-1322 terminal blocks, each with sixteen differential measurement channels. A RTD is conditioned by applying a small excitation current through its terminals, then measuring the voltage across the terminals and calculating the RTD resistance, which is pre-calibrated as a function of temperature. The chosen sensors have nominal resistance of 100 ohms at 0 °C and fit the standard European RTD polynomial curve (α=0.00385). The RTD's have an operating range from -50 °C to 204 °C, accurate to within ±0.45 °C, and with resolution (dictated by the input voltage limits and the 16-bit nature of the DAQ card) of 0.0078 °C [3]. Temperatures were taken from each sensor once each minute, recorded using National Instruments Labview software, logged into a standard tab-delimited text file, and analyzed using Microsoft Excel.

Thirty-six RTD’s were mounted to the stack structure, six on each of the non-heated tubes, and eight on each of the heated tubes. On the heated tubes, one sensor was placed directly on the heat source, and two neighboring sensors were placed 1” to the left and right of the source. The remaining sensors were placed at equally-spaced 60 degree circumferential intervals, named and located as shown below. The RTD’s were
mounted in equivalent positions on the single-tube structure. The mounting locations and adopted nomenclature are given in Figure 5.5.

Additionally, seven RTD's were mounted to the central column; four equally spaced vertically in a single line up the column, and four equally spaced circumferentially just below the interferometer mount (one RTD shared between both sets of four). Eight RTD's were placed to monitor the air temperature near the structure, four sets of two each at 90-degree placements, 4” from the surface of the structure. Each set was hung from a PVC pipe support, with one at the level of the gap between tubes 1 and 2 and one at the level of the gap between tubes 4 and 5. This was intended to capture the circumferential and vertical (heat rises) heating phenomena of the air, and validate through this trend that the tube sensors predominately measure the metal temperature.

One RTD was placed outside the isolation chamber to monitor the temperature fluctuations within the test area. All RTD's mounted to metal were contacted to a dab of thermally-conductive paste, secured to the surface using Kapton tape, and insulated from the air using a small section of 1-1/4" X 1/2” standard household foam tape. The RTD's measuring air temperature were hung freely from their supports.

5.3.1.2 Dimensional Metrology
Angular drift of the stack was measured using a Sago differential plane mirror interferometer (DPMI), mounted to a 5” diameter aluminum column placed within the stack assembly, and measuring the tilt of a reference mirror mounted to a horizontal reference plate attached to the top of the tube set. The optics were placed as shown in Figures 5.6 and 5.7. The interferometer laser was located outside of the thermal isolation chamber, and the 6 mm diameter beam from the laser entered the chamber through a 1” diameter, 0.5” thick optical window. The beam then was bent 90 degrees by a fold mirror, traveling upward parallel to the axis of the reference column, and entering the interferometer. At that point, the beam was polarized and split sequentially twice, into two source measurement beams for taking differential linear measurements of the motion of the top plate, and two reference beams for comparison. These beams were processed into a single optical signal, directed out the back of the interferometer to a fiberoptic pickup mount, and into a flexible fiberoptic cable routed to the measurement board. The ZMI 1000 measurement board connected to a Windows PC running the ZMI 1000 software, and the interferometer output was displayed on the screen and recorded to a text file as an angular value, in arcseconds. Single-point values were saved to disk once each minute during testing.
5.3.1.3 Application of Thermal Disturbances

Thermal disturbances were applied to the structure using Minco copper thin film heating elements, powered by a Xantrex XT 30-2 regulated DC power supply with a maximum nominal output of 60 W. Each source measured 1/2” x 1/2”, with a maximum operating temperature of approximately 80 °C.

For the initial tests, three sources were mounted directly to the structure, one each at the vertical mid-points of the three center tubes, parallel to the axis of the interferometer. This placement maximized the ability of the single-axis interferometer to capture the gross error motion from thermal expansion, as ideally the maximum differential expansion of the stack was seen by the measurement beams. The three sources were wired in a simple parallel circuit, so each was supplied the same voltage. The voltage across the circuit was monitored continuously throughout the test, and the power was calculated directly from
knowledge of the resistance across each heat source. Nominally, 3W (5V, 0.6A) was applied to the circuit, such that each source emitted nearly 1W.

5.3.1.4 Thermal Isolation Chamber

In order to isolate the structure from temperature disturbances, which can occur due to heating, cooling and air currents in the room, the test apparatus was placed in an insulated chamber. The chamber, with internal dimensions of 24” x 24” x 30” (L x W x H), was constructed of 4” thick extruded polystyrene (Styrofoam) with an R-value of 20 ($k = 0.029$ W/m K). The walls and top of the chamber were butt-jointed together and bonded using hot melt glue. All joints were sealed with Reflectix tape and silicone sealant. The seal along the bottom of the chamber was achieved by placing the chamber on a perimeter of 1.5” thick open cell neoprene, which deformed under the weight of the chamber. A two-inch thick layer of polystyrene insulated the exposed area of the table around the stack structure with a dimension of just under 24
in. X. 24 in. The stack itself was placed on top of a 1/16 in. thick layer of Buna-N rubber to isolate it from any thermal disturbances that were transmitted through the threaded holes in the optical table.

5.3.1.5 Test Procedure

For the initial tests using three contacting heat sources each emitting 1W, the following test procedure was established:

1. Seal the structure in the thermal chamber for at least four hours before testing, to allow any prior thermal non-uniformities from exposure to room air to attenuate.

2. Following this period, begin acquisition of interferometer and temperature data. Take a single-point reading from the Zygo measurement board once each minute, and cycle through all temperature sensors once each minute. Each cycle of temperature measurements consisted of a single-point reading from each sensor, with subsequent readings separated by a one-second delay.

3. After one hour of such data acquisition, activate the heat sources. Continue data acquisition with heating for the next six hours.
4. After these six hours of heating, deactivate the heat sources and continue data acquisition for the next hour.

5. After this one hour, terminate data acquisition and remove the thermal chamber to speed dissipation of heat from the structure in preparation for the next test. Leave the structure uncovered for at least four hours.

Hence, this '1/6/1' test specifies eight hours of continuous data acquisition, with one-hour non-heated periods preceding and following a six-hour heated period.

5.3.2 Finite Element Simulation Setup and Procedure

Prior to conducting the physical thermal stability experiments, a performance comparison between the segmented and one-piece control structures was made using finite element models in Pro/MECHANICA™. The structural solid models were simplified for simulation to those shown in Figures, 5.11 and 5.12, treating the tube sections as the only significant bodies, and replacing the kinematic coupling balls and grooves on the segmented structure with sets of 1.4” square (a very liberal estimate given the true nearly point contact) contacts between the nearly 1/8” (3 mm) tube-to-tube gaps.

Consistent with the laboratory test procedure, three 1/2” square heat sources were placed in a column at the horizontal centerlines of the second, third, and fourth segments, and at equivalent positions on the one-piece structure. Shown in red, these surfaces were defined as heat loads with input of 1W each. Using the thermobuoyant convection relations in Section 5.2.2, a uniform steady-state loss coefficient of 2 m-K/W was defined on the external cylindrical surfaces of the model, balancing the heat input of 3 W.

After building the solid model and defining the thermal loads and boundary condition, the thermally-induced deformation of the structure was determined by running consecutive Pro/MECHANICA™ thermal and structural analyses, using the results of the thermal analysis as the boundary condition for the structural analysis. Because this stepped analysis is unique, it is worthwhile to detail the simulation procedure:

1. Within the thermal module of Pro/MECHANICA™, the thermal simulation was executed using the aforementioned heat load and convection boundary conditions. A multi-pass adaptive analysis was defined, with 8th-order maximum convergence to within 10% of global and local energy norms. Given the heat source magnitudes, this convergence metric ensured the steady-state results exhibited a mature gradient.
2. After the thermal analysis completed, the solid model was opened in the Structure module of Pro/MECHANICA.

3. The bottom ring face was constrained in all six degrees of freedom.

4. A temperature load of type MEC/T Temp was defined, specifying the Analysis input as the thermal analysis of the structure, and the Load Set as the 3 X 1W load set. The option to Use Previous Design Study was not chosen, as it was found necessary to have Pro/MECHANICA generate a new mesh for the structural analysis rather than use the thermal analysis mesh from before. Consistent with the thermal analysis, the reference temperature was left to the default of zero.

5. The structural simulation was executed, defining the same analysis parameters as described in step 1 for the thermal simulation.

Each simulation converged within thirty minutes, using a 850 MHz Pentium III notebook, with 250 of the total of 512 MB system RAM allocated to the solver. Solution for the segmented model took approximately 75% longer than for the one-piece model.
5.3.3 Results

5.3.3.1 Finite Element Simulations

Having executed combined thermal and structural simulations, Figures 5.13 and 5.14 depict the solved steady-state constant temperature contours for the segmented and one-piece simplified models. Here, the hypotheses advanced in Section 5.2.2 are corroborated, with the heat disturbances directed prominently in the circumferential direction on the segments, while nearly equally in the axial radial directions on the one-piece structure. Within the range and resolution displayed, the contours remain elliptical on the one-piece structure, and approach the suggested hyperbolic form near the unheated side of the segmented structure. Significant temperature change within this resolution propagates nearly fully (180 degrees) around the segments, yet no more than 60 degrees around the tall single piece structure. The unheated segments show near perfect uniformity, being well-isolated from the disturbances by the stainless steel kinematic couplings.

Figure 5.13: Steady-state temperature contours on segmented model.
Figures 5.15 and 5.16 show the axial displacement distribution on the structures. In both cases, the axial displacement of the top surface of the assembly is slightly greater on the heated side than on the non-heated side, and this difference is greater for the one-piece structure. Specifically, the top of the heated face of the segmented structure displaces 22.3 microns, and the top of the non-heated face displaces 21.7 microns, equaling a uniaxial tilt of 0.46 arcseconds. Conversely, the top of the heated face of the one-piece structure extends 20.7 microns, while the non-heated face extends 19.8 microns, equaling a uniaxial tilt of 0.70 arcseconds. Thus, by simulation, the one-piece structure is 52% more thermally sensitive than the segmented design.
It is also important to consider the thermally-induced deformation in the radial direction. While uniform radial expansion would keep the axes aligned perfectly, consistent with kinematic coupling theory, non-uniform expansion would lead to translation at the sample position of no less than one third the magnitude of the non-uniformity. However, assuming excess of static friction between the coupling ball and groove at a particular joint, the advantage of the coupling is manifested in the ability for relative motion between segments, hence keeping their central axes aligned while also thermally isolating them. Capability for relative motion was not built into the structural models presented, but differences in radial expansion at the outer surfaces were queried to be as much as 0.1 micron, approximately 20% of the equivalent objective plane deformation caused by the asymmetric axial expansion.
5.3.3.2 Laboratory Experiments

Three sets of measurements were taken on each of the two structures, following the 1/6/1 hour schedule of heating with three 1 W sources described in Section 5.3.1.5. This section presents the angular interferometer measurements and selected temperature measurements for one trial on each structure; the metrics reported were repeatable within 0.1 arcsecond and 0.01 °C between the respective sets of trials.

Each plot is accompanied by a short explanation. In advance summary, the results demonstrate:

1. Maximum angular deflection of 0.6 arcsec for the segmented structure and 1.0 arcsec for the one-piece structure, translating to linear drift of approximately 675 and 1090 nm at the central objective position. These values represent a 60% performance degradation, or vice-versa a 40% performance improvement, between the segmented and one-piece designs.

2. Circumferential temperature differences of 0.10-0.25 °C across the heated tubes of the segmented structure, which are nearly constant over the heated duration of the experiments. Differences across all levels of the one-piece structure are 0.05-0.13 °C.

3. Nearly perfect circumferential temperature uniformity in the chamber air, around the non-heated tubes, and around the central column reference. Hence, the tubes act as thermally isolated bodies, and the interferometer readings are not subject to error from angular drift of the central column.

These results are largely consistent with the Pro/MECHANICA simulations; specific comparisons are given in the next section.
The interferometer values shown in Figures 5.17 and 5.18 are taken as 30-minute (30-sample) moving averages, which smooth the 0.06 arcsec resolution of the differential measurement. The steeper gradients are seen at the commencement (at $t = 60$ min) and termination (at $t = 360$ min) of heating. Maximum drifts of 0.60 arcsec and 0.87 arcsec are seen respectively for the segmented and one-piece structures, corresponding to linear movements of approximately 725 nm and 1050 nm at the central objective position. When the disturbance is applied, the deflection rapidly moves to an intermediate value as heat is reaching the sensors on the heated side before it flows around the circumference and affects the sensors on the non-heated side. When the sensors on the non-heated side are disturbed - when the iso-temperature contours drawing in Figure 5.4 are advancing at nearly the same rate at the heated and non-heated sensor positions - the rate of increasing deflection stabilizes. When the disturbance terminates, analogous rapid relaxation occurs, and then the relaxation slows and trends toward zero.

Figure 5.17: Deflection of segmented structure versus time (measurement duration extended to 13 hours to show thermal relaxation).
Comparing the transient profiles of the two structures as in Figure 5.18, the response of the one-piece structure during the first hour of heating is greater than the segmented structure; 120 minutes after measurement starts, the one-piece structure has drifted approximately 0.7 arcsec, while the segmented structure has drifted approximately 0.3 arcsec. This is a much greater discrepancy than after 8 hours of measurement, as after the initial heating period roles reverse and the one-piece structure continues to drift at a considerably slower pace than the segmented structure. This can be rationalized in terms of the ratios of axial to radial dimensions of the structures. Upon application of heat, the axial flow in the one-piece structure is continuous over a greater distance than in the segmented structure, hence more heat is directed circumferentially around the segments than around the one-piece tube. This shows that the segmented structure is far
superior in disturbance rejection, being 60% less sensitive to thermal input than the one-piece structure
during the first hour of heating. Although the performance gap closes to only a 30% advantage over the
course of several hours, the microscope design seeks stability of the image spot over approximately one
hour, making the initial transient performance most important.

Several sets of interferometer data were also taken under identical experimental conditions, only with-
out heat applied to the structure. These results verify the stability of the interferometer readings within 0.12
arcsec over several hours, subject to noise equal to the resolution magnitude of 0.06 arcsec.

Figure 5.19: Normalized temperatures (10-minute averages) around tube 1 (non-heated) of segmented
structure.
Figures 5.19 and 5.20 show how the non-heated tubes of the segmented structure maintained almost perfect circumferential symmetry of temperature throughout the experiments; hence the air gaps and spot kinematic contacts between the tubes are excellent thermal isolators. The tubes heated uniformly by approximately 0.7 °C during the test duration, yet the end-to-end range remained stable and centered at zero with RMS deviation of 0.02 °C.

Figure 5.20: End-to-end normalized temperature difference on tube 1 (non-heated) of segmented structure.
Figure 5.21 displays the delayed circumferential heating pattern of the heated segments, as the sensors near the heat source show initial temperature changes approximately ten minutes after the start of heating, and the heat then flows around the tube. The farthest sensor is activated approximately ten minutes after the start of heating, after which all sensor temperatures increase at nearly constant and identical rates until heating is discontinued.

Figure 5.21: Normalized temperatures (10-minute average) around tube 3 (heated) of segmented structure.
Ten minutes after heating started, the end-to-end temperature range increased to 0.22 °C, gradually increased to 0.24 °C, and then gradually decreased until heating ends. This gradual decline is a motion toward a long-term steady-state circumferential gradient, predicted by simple calculations to be achieved after approximately 20 hours of constant heating.

Figure 5.22: End-to-end normalized temperature difference on tube 3 (heated) of segmented structure.
Figure 5.23 emphasizes the constancy of shape in the circumferential profiles starting approximately thirty minutes after heating begins. Once the steady-state gradient is established, relative temperature changes are uniform around the tube.
Figure 5.24: Normalized temperatures (10-minute average) around level 3 of one-piece structure.
Figures 5.24 and 5.25 show the characteristic thermal evolution of the levels of the one-piece structure. Trends for the third level are shown in Figure 5.25 with a steady non-uniformity of 0.12 °C. The behavior of all levels of the one-piece structure is similar to that of the heated tubes of the segmented structure, as the heat spreads radially from the sources and is not constrained in the axial direction until it flows to the absolute ends of the structure. Because of the lack of axial constraint, the total circumferential heating at this level is 1.2 °C, compared to 1.6 °C for the third segment. Similarly, the 0.12 °C steady circumferential non-uniformity here is approximately half of the 0.23 °C value for the third segment. However, the one-piece structure carries such a difference fully along its length, whereas the segmented design restricts it to the heated segments. Because the segments channel the disturbances around the tubes, and prevent axial transmission between segments, the error motion of the segmented structure is significantly diminished.

Charts in Appendix C show that the air temperature inside the chamber was circumferentially uniform at both levels of measurement, and insensitive to fluctuations in the external laboratory temperature. The
air temperatures near and far from the heaters tracked almost perfectly upon application of the heat sources, confirming that the tube-mounted sensors were effectively recording the metal temperatures. Furthermore, the central reference column, to which the interferometer was mounted, exhibited uniform heating with circumferential temperature differences no greater than 0.02 °C, significantly less (less than approximately 0.05 arcsec) than the non-uniformity around the structure. Therefore, expansion of the central column had a negligible effect on the interferometer readings of the structural tilt.

5.3.3.3 Comparison - Validation of Simulation Models

To this point, qualitative similarities between the simulation results and the experimental results are evident; as predicted by Pro/MECHANICA, the non-heated segments exhibited near-perfect circumferential uniformity throughout the tests, while the heated segments and all levels of the one-piece structure showed considerable steady-state temperature differences. Quantitatively, Table 5.2 compares the temperature differences (heated side minus non-heated side) at each tube centerline level for both structures, when simulated using Pro/MECHANICA and when measured in the laboratory. At the heated levels, values are averages of those from the two sensors placed 1” from the sources.

<table>
<thead>
<tr>
<th>Level (1 = bottom)</th>
<th>ΔT Segmented - Simulated</th>
<th>ΔT Segmented - Measured</th>
<th>ΔT One-Piece - Simulated</th>
<th>ΔT One-Piece - Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.01</td>
<td>0.00 +/- 0.01</td>
<td>0.07</td>
<td>0.06 +/- 0.01</td>
</tr>
<tr>
<td>2</td>
<td>0.12</td>
<td>0.13 +/- 0.02</td>
<td>0.12</td>
<td>0.09 +/- 0.02</td>
</tr>
<tr>
<td>3</td>
<td>0.18</td>
<td>0.21 +/- 0.03</td>
<td>0.12</td>
<td>0.12 +/- 0.01</td>
</tr>
<tr>
<td>4</td>
<td>0.12</td>
<td>0.12 +/- 0.02</td>
<td>0.12</td>
<td>0.09 +/- 0.02</td>
</tr>
<tr>
<td>5</td>
<td>0.01</td>
<td>0.00 +/- 0.01</td>
<td>0.07</td>
<td>0.06 +/- 0.01</td>
</tr>
</tbody>
</table>

Table 5.2: Simulated and measured end-to-end circumferential temperature differences [°C].

Within the levels of uncertainty (the 2-sigma confidence intervals between trials) of the experiments, the simulated and measured temperature differences are within 0.01 °C at all levels of both structures. The simulation reports steady-state results after at least 20 hours of heating, while the experimental measurements are transient values after only 4-6 hours of heating. However, with exponential transient progression
of heat, the temperature difference across a solid body subject to a constant heat input is nearly stable soon after application of the disturbance, as confirmed by the end-to-end difference progressions in Figures 5.21 and 5.24. Furthermore, the simulation model assumes that the coupling contacts are the only heat flow between the segments; since for simplicity no boundary volume is defined around the segmented structure and no outflow condition is specified for the intermediate top and bottom faces of the tubes, there is no convection flow across these interfaces. The specification of a uniform convection coefficient across the entire outer surface of the structures, which clearly inaccurately assumes uniform temperature distributions, is another small, yet permissible discrepancy.

Beyond these reasons, the true goal of this study is an ordinal optimization in two respects: first to establish the approximate advantage of the segmented design over the one-piece design; and second to ordinally determine the best geometry for the segments. Especially in the second case, the absolute performance values are less important than the relative rankings of the candidate designs.

5.3.3.3 Analytical Model of Thermal Expansion

It is also instructive to compare the time-varying laboratory interferometer measurements to predictions of the thermal expansion of the segmented structure. Using the thirty-six temperature measurements from the surfaces of the segmented structure as inputs, two predictions of the deflection were made: one assuming a constant vertical temperature distribution on each segment; and the other using a conventional transient approximation for spatial and temporal deviation. The two predicted trends and the measured trend are shown in Figure 5.26.

The first prediction is a good approximation of the steady-state tilt of the structure, yet fails to capture the behavior of gradual expansion which precedes the steady state. In assuming constant vertical temperature profiles on each segment (taking the RTD measured value at each location), this method ignores the transient delay associated with heat flow in the vertical direction. By using the direct measurements along the horizontal centerlines of the tubes the delay in circumferential flow is directly accounted for, yet the results over-estimate the deflection due to vertical heat flow until the vertical temperature profiles are truly constant at a near steady-state at $t = 350$ minutes. This estimate is based on a simple finite sum of the ther-
mal expansions of each segment, differenced between the heated and non-heated sides. The difference in linear expansion is calculated as:

$$\delta = \alpha \left[ \sum_{i=1}^{5} L_i T_i \right]_{heated} - \left[ \sum_{i=1}^{5} L_i T_i \right]_{nonheated}$$  \hspace{1cm} (5.17)

where \(L_i\) is the length of each segment, and \(T_i\) is the measured temperature at its horizontal centerline. From this linear difference, the angle is found using Equation 5.3. In providing a good steady-state estimate, this relation can be useful in macroscopically comparing designs when only the surface temperatures are recorded.

The second method is a good prediction of the shape of the transient deflection profile, yet overestimates the magnitude of the deflection by as much as a factor of two. In this method, the vertical temperature profiles in the segments take the shape of the depthwise profiles in a semi-infinite solid body, given by the relations [2]:

$$T_{norm} = \frac{T(x, t) - T(0)}{T_s - T(0)} \left( 1 - erf \left( \frac{x}{2 \sqrt{\alpha t}} \right) \right)$$  \hspace{1cm} (5.18)

$$erf(\eta) = \frac{2}{\sqrt{\pi}} \int_0^{\eta} e^{-z^2} dz$$  \hspace{1cm} (5.19)
In Equation 5.18, the heated surface temperature, $T_s$, is invariant with time. To adapt the experimental data to this relation, $T_s$ was specified as time-varying, with an instantaneous value equal to the temporal average of the measured value to that time. For ease of computation in a spreadsheet, the averages were updated incrementally with each time step [4]:

$$
\bar{T}_{s,n} = \left(\frac{n-1}{n}\right)T_{s,n-1} + \frac{T_{s,n}}{n},
\tag{5.20}
$$

where $T_{s,n}$ is the instantaneous measured value along the horizontal centerline of each segment at time instance $n$. To make computation manageable, each segment was discretized into five piecewise constant sections, over which the transient profile was superimposed. This method of averaging roughly accounts for the latency in vertical heat flow due to changes in the disturbance temperature, $T_s$.

The semi-infinite body transient approximation is definitely crude, but is by far the simplest transient relationship available in closed form. A more appropriate closed-form solution could be one for a finite

![Figure 5.26: Comparison of measured and predicted angular deflections of segmented structure.](image)
prismatic solid body; however, that solution is based on an infinite series, and the increased complexity would not justify the slightly better (seen from $Fo = 0.08$ for semi-infinite body vs. $Fo = 0.07$ for finite body) solution. In either case, these methods are one-dimensional solutions, ignoring that the real case is one of two-dimensional conduction (vertical and circumferential) along the surface from the thin-film sources used in the tests. Furthermore, these methods assume that the heat source is uniform and time-averaged over the disturbed surface, while really it is localized and instantaneously variant.

A more accurate method would be to treat each segment as a series of slices, with each slice a finite thin volume with lumped (constant temperature) capacitance. However, this is approaching the finite element method, which enables appropriate assignment of boundary conditions and localized heat sources. Knowing the accuracy of the steady-state simulations from Pro/MECHANICA, similar transient simulations would be the best way of estimating the time-variant deflection absent direct experimental measurements.

5.3.4 Design Optimization

Validation of the finite element model for the test configurations permitted its use for design optimization with respect to tube materials, tube thickness, and the number of structural segments. First, with the prototype dimensions, a performance comparison between the materials listed in Table 5.1 was made. Second, choosing the optimal material from this analysis and keeping the total length and inside diameter of the structure fixed (both packaging constraints), tube thickness and the number of segments were varied within reasonable bounds.

<table>
<thead>
<tr>
<th>Material</th>
<th>Average $\Delta T$ [$^\circ$C] - segmented</th>
<th>Tilt [arcsec] - segmented</th>
<th>Average $\Delta T$ [$^\circ$C] - one-piece</th>
<th>Tilt [arcsec] - one-piece</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum (6061-T651)</td>
<td>0.08</td>
<td>0.46</td>
<td>0.10</td>
<td>0.70</td>
</tr>
<tr>
<td>Copper</td>
<td>0.04</td>
<td>0.16</td>
<td>0.05</td>
<td>0.23</td>
</tr>
<tr>
<td>Brass</td>
<td>0.13</td>
<td>0.64</td>
<td>0.14</td>
<td>0.90</td>
</tr>
<tr>
<td>Stainless Steel (AISI 410)</td>
<td>0.59</td>
<td>1.93</td>
<td>0.60</td>
<td>2.62</td>
</tr>
</tbody>
</table>

Table 5.3: Tube material optimization study results ($\Delta T =$ heated minus non-heated).
Table 5.3 gives the results of the tube material optimization study, showing that the error motion of a copper tube segmented structure is 0.16 arcsec, versus 0.46 arcsec for the prior simulated aluminum tube structure, a reduction of 65%. Among the four materials studied, copper is best, aluminum is second best, brass is third best, and stainless steel is in a distant fourth place. For all materials, the ratio of tilt of the segmented structure to tilt of the one-piece structure is approximately constant at 1.45. The axial displacement contours for segmented copper and fully stainless steel structures are contrasted by Figure 5.27 and 5.28, showing a much greater angular nonuniformity in the stainless steel model.

Hence, in applications such as single molecule experiments where material and manufacturing costs are less important design variables than thermal performance, pure copper, or a copper alloy with a transient performance index value greater than that of aluminum, should definitely be chosen. For a low-cost structure with high thermal performance, for which strength and stiffness are not most critical, the aluminum segmented design should be elected. When strength requirements mandate a steel structure, placing
air gaps between modules will provide at least a 30% performance improvement in terms of the error motion studied here. In this case, more novel alternatives such as insulating the outside of the structure or placing a high-conductivity shield over the low conductivity steel (with a gap between the two to prevent transfer of thermal strains) can be studied.

Having optimized the material choice among four candidates, next a geometric design study was performed, with copper as the material choice. Keeping the total tube length and the inside diameter as packaging constraints, the tube thickness of both structures was varied within reasonable bounds of 1.0” and 2.5”. Next, for the segmented structure, total length, inside diameter, and thickness were held constant and the number of segments was varied. For simplicity of the analysis, one 3W heat source was applied at the horizontal centerline of the structure during the geometric optimization studies. The total length, including the kinematic couplings between segments, was shortened to 500.0 mm from 508.6 mm, and the segments were all equal in length. In this model, the segment length [mm] was calculated using:

\[ L_s = \frac{500 - 3(n - 1)}{n}, \]  

(5.21)

where \( n \) is the number of segments and 3 mm is the gap height between the segments.

Figure 5.29 shows that the error motion monotonically decreases with increasing tube thickness for the one-piece structure and for the structure with five segments. Hence, a better design is one with five thick segments, and a structure with five segments always outperforms a one-piece structure of the same thickness. The data series marked by triangles shows the error motion with thickness fixed at 1.5”, with the number of segments varying (in increments of two) from one to nine. The relationship here is better shown in Figure 5.30, indicating that the error motion is minimized with five segments, and increases in both bounding directions. When the results with varying thickness and with varying number of segments are collapsed onto the same plot in terms of the segment height to thickness ratio, Figure 5.31 results. Here, the (green and blue) curves for when thickness is varied are similar in shape (within the convergence confidence of the simulation), and are simple translations along the black curve for when the number of segments is varied.
Figure 5.29: Angular deflection of segmented and one-piece copper structures with varying thickness.

Figure 5.30: Angular deflection of structure with varying number of segments.
It was not intuitive that the error motion would decrease with increasing thickness; however, it is reasoned that a thicker tube presents greater thermal capacitance to absorb the constant disturbance, therefore decreasing the overall temperature change of the structure. The thermal expansion is directly related to the temperature difference between the heated and non-heated sides, which is a function of the magnitude of the overall temperature change, and the ease with which heat flows in the circumferential direction relative to the axial direction. Constraining the axial flow using a shorter segment is advantageous only until the effect of the decreased thermal capacitance of the segment takes over, after which the greater overall heating of the segment increases the magnitude of the thermal expansion.

The final iteration was to model a structure with uniform layers of foam insulation (k = 0.026 W/m-K) bonded to the outside segment surfaces, and apply the thermal disturbances. Figure 5.32 shows the insulated model of the segmented structure with 1.5” thick bands of insulation constrained to the 2.5” thick
copper tube segments. Individual 1W total flux heat sources were applied at the usual locations of the second, third, and fourth segments. The insulation acts as an excellent primary dissipator; not only does the thermal mass of the insulation absorb some of the disturbances by bulk heating, but by the time the temperature gradient reaches the outer metal surfaces, it is significantly more distributed (both circumferentially and axially) than when directly applied.

Hence, within the bounds tested, the best design is a five-segment structure with copper metal tube segment thickness of 2.5”, covered with a 1.5” thickness of foam insulation. With three 1W heat sources applied as before, this structure has simulated angular deflection of 0.04 arcsec, a 93% reduction from the 0.46 arcsec value for the 1.5” thick prototype aluminum structure. While there is confidence in the results of this optimal design study, given more time it would be useful to evaluate robustness by testing a varying number of segments at thicknesses other than 1.5”, and with varying heat source magnitudes and distribu-
tions. It also seems like a more unified analysis could be achieved by normalizing parameters in terms of the thermal capacitance of the rings, rather than just the dimensions of segment length and thickness.

5.4 Future Work and Conclusions

While optimization through iterative finite element simulation significantly enhanced the performance of the design recommendation, additional bench-level thermal experiments would also be useful. Specifically, it would be instructive to assess performance with a layer of foam or fiberglass insulation mounted to the outside tube surfaces and relate the results to the simulations with insulation presented before.

Furthermore, in these experiments, the limited resolution of the interferometer necessitated that relatively large thermal disturbances be applied to generate well-measured deflection trends. More realistic thermal performance measurements, using lesser disturbances or exposing the structure directly to a well-controlled ambient, could be made by using high-precision capacitance probes mounted on the central reference, and measuring the deflection of the top of the structure. With three capacitive sensor pairs mounted vertically, one could sense biaxial rotation and axial extension of the top reference. With two to four additional sensor pairs mounted radially, one could sense in-plane drift of the structure. Capacitive sensors are available with linear resolution of less than 1 nm; sensors of this capability could improve the resolution of angular measurements by a factor of 60.

This work presented and validated the concept of a segmented, modular tube, kinematically-coupled structure as applied to a high-precision microscope for single molecule experiments. The structure was shown to be significantly less sensitive to asymmetric thermal disturbances than a single-piece tube alternative, and the kinematic interfaces between the segments offer significant functionality in disassembly, reconfiguration, and reassembly of the system without need for re-calibration.

More generally, these results demonstrate the feasibility of the segmented design for modular serial assemblies, in instrumentation structures including microscopes and high-precision reconfigurable measurement equipment such as large coordinate measuring machines, and in machine structures such as the industrial robot studied in the previous chapter. For robots, modules can be removable, perhaps wirelessly presence sensed and controlled, motor-driven axes. With different modules, the manipulator can easily be reconfigured to multiple end-effectors, extended reach kits, and even different numbers and types of joints.
Fixed-base parallel manipulators can be built from elementary structural sections similar to those of the segmented microscope structure, with links mounted to the outside tube surfaces extending parallel to ground. Results of the forthcoming repeatability and exchangeability experiments on the segmented prototype will quantify the mechanical modularity performance of such a design.