*Thermal Issues Associated with the HVAC
and Lighting Systems Influences on the
Performance of the National Ignition Facility
Beam Transport Tubes*

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by

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Abstract

This report summarizes an investigation of the thermal issues related to the National Ignition Facility. In particular, the influences of the HVAC system and lighting fixtures on the operational performance of the laser guide beam tubes are reviewed and discussed. An analytical model of the oscillating HVAC air temperatures in the NIF switchyard and target bay will cause significant amounts of beam distortion. However, these negative effects can be drastically reduced by adding thermal insulation to the outside of the beam tubes. A computational fluid dynamics model and an analytical investigation found that the light-fixture to beam-tube separation distance must be on the order of 5.7 m (18.7 ft) to maintain acceptable beam operating performance in the current NIF design. By reducing the fluorescent light fixture power by 33% this separation distance can be reduced to 3.5 m (11.5 ft). If in addition, thermal insulation with a reflective aluminum foil covering is added to the outside of the beam tubes, the separation distance can be reduced further to 1.6m (5.2 ft). A 1.27 cm (0.5 in.) rigid foam insulation sheet with aluminum foil covering will provide adequate insulation for the beam tubes in the NIF switchyards and target bay. The material cost for this amount of insulation would be roughly \$30,000.

NOMENCLATURE

Symbol	Description
A	Area
c_p	Specific heat at constant pressure
d	Diameter
E	Total hemispherical emissive power
F	Radiation exchange view factor
g	Gravitational constant (9.18 m/s^2)
h	Heat transfer coefficient
J	Radiosity
k	Thermal conductivity
L	Beam length or characteristic length
l	Characteristic length
P	Period of oscillation, perimeter
Pr	Prandtl number ($c_p \mu/k$)
q	Heat transfer rate
Ra	Rayleigh number
Re	Reynolds number ($\rho u l/\mu$)
S	Beam width
T	Temperature
t	Time
U	Overall heat transfer coefficient
V	Volume
w	beam tube or light fixture width
Greek Symbol	Description
α	Thermal diffusivity
$\Delta\theta$	Beam pointing error
ΔT	Temperature difference
ε	Emissivity
μ	Dynamic viscosity
ρ	Density
σ	Stefan-Boltzmann constant ($5.670 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$)
ν	Kinematic viscosity
Subscript	Description
air	Air
amp	Amplitude of oscillation
b	Black body
$conv$	Convection heat transfer

<i>env</i>	Environment
<i>in</i>	Insulation
<i>L</i>	Parameter corresponding to characteristic length <i>L</i>
<i>max</i>	Maximum
<i>mean</i>	Mean value
<i>mh</i>	Light fixture metal housing
<i>pc</i>	Light fixture plastic cover
<i>rad</i>	Radiation heat transfer
<i>s</i>	Surface
∞	Property corresponding to far field environment

1. Introduction

This report summarizes an investigation of the thermal issues related to the National Ignition Facility. In particular, the influences of the HVAC system and lighting fixtures on the operational performance of the laser guide beam tubes are reviewed and discussed. It has been recognized that excursions in the NIF environmental air temperature from a mean operating value and imposed heat loads from lighting fixtures may significantly influence the temperature and hence the operating performance of the NIF laser beam transport tubes.

Nonuniform gas temperatures across a beam tube (perpendicular to the beam direction) cause a gradient in the gas index of refraction which results in laser pointing errors. A static temperature gradient that does not change over a period on the order of one hour, such as that imposed by waste heat from light fixtures, would lead to pointing errors that could be corrected by a beam alignment system. However, uncorrectable changes in gas temperature gradients over shorter time periods, such as those imposed by the oscillating HVAC system, would result in undesirable beam pointing errors. John Murray [1] showed in a previous memo that the laser beam steering, $\Delta\theta$, in an argon atmosphere, for a linear temperature profile across the beam tube in a direction perpendicular to the beam propagation, can be represented by

$$\Delta\theta = 0.89 L (\Delta T / S) [\mu\text{rad}], \quad (1)$$

where L [m] is the beam length, S [m] is the beam width, and ΔT [K] is the steady-state temperature difference across the beam tube.

The purpose of this report is to assess the influence of HVAC air temperature excursions and waste heat from light fixtures on the temperature gradient and associated beam pointing errors of the NIF laser beam transport system. Two different studies are discussed in this report. First, an analytical investigation of the NIF environment air temperature fluctuations caused by the HVAC control system and the corresponding effects on the beam pointing error are reviewed. A parameter study summarizes the effects of the oscillating air temperature (amplitude and period) and beam tube insulation thickness on beam pointing errors. Next, experimental temperature measurements and a discussion of the heat transfer characteristics of fluorescent light fixtures are presented. In addition, a detailed numerical study of the development of thermal plumes from light fixtures was performed using the commercial computational fluid dynamics software package CFX. The extent of the thermal plume and its influence on beam tube location relative to a light fixture are discussed. Finally, an analytical investigation of the thermal radiation interaction between a light fixture and beam tube is used to determine allowable light-fixture-to-beam tube separation distances required to meet acceptable operating performance criteria. These studies as a whole provide the design criteria for HVAC and lighting systems that are necessary to avoid interference with the operating performance of the NIF.

2. HVAC System Evaluation

2.1 Beam Tube Wall Temperature Response to HVAC System

Because of practical accuracy limitations in monitoring probes and controllers, the continuously operating NIF HVAC system is expected to provide a continuously fluctuating air supply temperature about a mean operating value. In addition, the complex geometry of the structures in the switchyard will undoubtedly produce in nonuniform flow patterns (and corresponding nonuniform heat transfer coefficients) around the periphery of the beam tubes. The authors observed and measured highly nonuniform velocities around beam tubes in the NOVA facility (at Lawrence Livermore National Laboratory) which is a predecessor to NIF. Consequently, the four surfaces of the beam tubes will respond differently to changes in the surrounding air temperature. This nonuniform temperature response will result in time fluctuating temperature gradients across the beam tube and thus lead to beam pointing errors. The effect of a nonuniform temperature response of the beam tube surfaces is displayed in the following model.

It is assumed that the NIF environmental air temperature, T_{air} , follows a periodic function given by

$$T_{air}(t) = T_{mean} + T_{amp} \sin(2\pi t / P), \quad (2)$$

where T_{mean} is a mean air temperature, T_{amp} is the amplitude of the temperature fluctuation, t is time, and P is the period of oscillation.

A simplified one-dimensional transient energy balance on a single side of the beam tube, modeled as an isothermal surface, gives (neglecting radiation heat transfer)

$$hA (T_s - T_{air}) = \rho V c_p dT_s/dt, \quad (3)$$

where h is a heat transfer coefficient, and T_s , A , ρ , V , and c_p are the temperature, surface area, density, volume, and specific heat, respectively, of the beam tube material.

Substituting Equation (2) into Equation (3) and integrating give

$$T_s(t) = \frac{a_1 a_2}{1 + a_1^2 a_3^2} \left[a_3 e^{-t/a_1} + a_1^{-1} \sin(a_3 t) - a_3 \cos(a_3 t) \right], \quad (4)$$

where

$$a_1 = V \rho c_p / h A,$$

$$a_2 = T_{amp},$$

$$a_3 = 2 \pi / P.$$

The average heat transfer coefficient, h_L , can be evaluated with the following empirical Nusselt correlation for laminar external flow over a flat plate [2]

$$\bar{h}_L = 0.664 \frac{k}{L} \text{Re}_L^{1/2} \text{Pr}^{1/3}, \quad (5)$$

where k is the air thermal conductivity, L is the length of the surface parallel to the flow direction, Re is the Reynolds number, and Pr is the Prandtl number.

Using air properties at 300 K and 101.3 KPa and air velocities ranging from 0.05 m/s to 0.250 m/s (10 fpm to 50 fpm), Equation (5) gives the following corresponding range for the heat transfer coefficient

$$1.24 \leq h_L \leq 2.76 \text{ (W/m K)}.$$

The range of air velocities used in this study was based upon values reported in a previous computational fluid dynamics analysis of the NIF HVAC systems [3].

In this analysis, it was assumed that opposite faces of the beam tube may have heat transfer coefficients that differ significantly. For a worst-case scenario, a model was developed in which external isothermal faces on opposite sides of the beam tube exposed to convection heat transfer coefficients of $1 \text{ W m}^{-1} \text{ K}^{-1}$ and $5 \text{ W m}^{-1} \text{ K}^{-1}$. Next, Equation (4) was solved for both of these surfaces to generate transient temperature response plots such as those shown in Figure 1. The temperature-time histories of Figure 1 indicate that the sinusoidal fluctuations of the beam tube surface temperatures lag behind the oscillating air temperature. More importantly, Figure 1 displays how the temperature difference between the two beam tube surfaces oscillates between zero and a maximum value. These results are consistent with those presented previously in a similar HVAC study by Marshall et al. [4].

Figure 1. Oscillating HVAC air temperature and the corresponding temperature responses of beam tube surfaces with different heat transfer coefficients.

2.2 Beam Tube Gas Response to Beam Tube Wall Temperature Oscillation

To determine the effects of the temperature differential of the two opposing beam tube surfaces on the laser beam propagation, the response of the gas within the beam tube to the fluctuating wall temperatures must be known. It is the temperature distribution in the beam tube gas that leads to spatial variations in the index of refraction and the corresponding beam distortion effects.

As the beam tube wall temperatures rise and fall, heat will be transported to and from the gas in the beam tube by diffusion and advection. Assuming for the time being that diffusion is the primary transport of heat, an analytical model can be created which describes the temperature response of the beam tube gas to a fluctuating wall temperature. The following form of the heat diffusion equation describes the one-dimensional transient diffusion of heat between a wall with a periodic temperature and a gaseous medium,

$$\frac{\partial^2 T}{\partial y^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t}, \quad (6)$$

with the following boundary and initial conditions

$$T(y, 0) = 0$$

$$T(y \text{ approaches } \infty, t) < \infty$$

$$T(0, t) = T_{amp} \sin(2\pi t / P),$$

where T_{amp} and P are the amplitude and period of the beam tube wall temperature oscillation that come from solving Equation (4).

A solution to Equation (6) with the above boundary conditions is given by [5]

$$T(y,t) = T_{amp} \exp\left[\frac{-y}{\sqrt{\alpha P / \pi}}\right] \sin\left[\frac{2\pi t}{P} - \frac{-y}{\sqrt{\alpha P / \pi}}\right]. \quad (7)$$

Using an HVAC system temperature oscillation amplitude and period of 0.28°C and 15 minutes, respectively, Equation (4) was solved for the beam tube wall temperature response for an external heat transfer coefficient of 5 W/m² K. The chosen HVAC parameters are representative of the intended operating values for the NIF final design. This analysis was performed for a beam tube with and without 1.27 cm (0.5 in.) thick insulation with a thermal conductivity of 0.019 W/m K (see next section for discussion on insulation modeling). The resulting beam tube temperature amplitudes were roughly 0.0045°C and 0.0200°C for the models with and without insulation, respectively. Using these temperature amplitudes along with an oscillation period of 15 minutes, Equation (7) was solved to determine the corresponding beam tube gas temperature responses. The solutions to Equation (7) for the bare and insulated beam tube internal gas temperatures are displayed in Figures 2 and 3, respectively.

The transient beam tube wall temperature creates a corresponding transient temperature response in the gaseous medium. Figure 2(a) shows a three dimensional (spatial and temporal) plot of the gas temperature response for the noninsulated beam tube. Immediately evident in the temperature plot of Figure 2(a) is the temporal rise and

(a)

Figure 2. Noninsulated beam tube gas temperature response from an oscillating beam tube wall temperature with an amplitude of 0.02 degrees Centigrade and a period of 900 seconds: (a) 3-D plot and (b) temperature vs. distance plots for three different times.

(b)

(a)

(b)

Figure 3. Insulated beam tube gas temperature response from an oscillating beam tube wall temperature with an amplitude of 0.02 degrees Centigrade and a period of 900 seconds: (a) 3-D plot and (b) temperature vs. distance plots for three different times.

fall of the gas temperature in close proximity to the beam tube wall ($y < 0.1$ m). Also of interest is the limit of the spatial extent of the thermal disturbance into the gas medium. The oscillating temperature of the wall is not significantly communicated to the gas at a distance greater than roughly 0.2 m. This is further conveyed in the two-dimensional temperature plots of Figure 2(b) which show the gas temperature profile for three instances in time. These three plots show the lack of temperature response of the gas at a distance greater than 0.2 m from the wall. The relatively large temperature gradients near the wall would lead to significant distortion of the laser beam in that area while the beam in the central portion of the beam tube would be nearly undisturbed. The transient and nonlinear behavior of the temperature profile in the beam tube gas prevents a straightforward and accurate assessment of the resulting beam distortion effects. However, an indication of the effects of the temperature gradient can be assessed with the temperature data of Figure 2(b) for the time corresponding to 300 s. From this temperature versus distance plot, a linear temperature gradient of $-0.121^{\circ}\text{C}/\text{m}$ ($-0.017^{\circ}\text{C}/0.14$ m) can be approximated through the gas in the near vicinity of the wall (<0.14 m). Using this temperature gradient over a beam tube length of 40 m (representative of the NIF switchyard), Equation (1) predicts a beam pointing error of 4.3 μrad for the outer segment of the beam nearest the wall. Since the remaining interior segment of the beam would be relatively undisturbed, the net effect would be a significant distorting of the beam spot size at the final target.

The beneficial effects of beam tube insulation are shown in the gas temperature response plots of Figure 3. For comparison purposes, the same scales have been used in Figure 3 as in Figure 2. The three dimensional temperature plots of Figures 3(a) and 3(b)

show significantly less magnitude and spatial variation in the beam tube gas temperature as compared to the uninsulated beam tube case of Figure 2. A beam distortion analysis similar to that which was done for the uninsulated beam tube predicts a beam pointing error of only 1.2 μ rad for the outer portion of the beam in the insulated beam tube. From this comparison, it is obvious that the net result of the beam tube insulation is significantly less beam distortion both on a local and global basis.

Figures 4(a) and 4(b) display, respectively, the cross-section of a quad laser beam tube for the NIF switchyard beam transport line, with and without insulation. Based upon the temperature response plots of Figures 2 and 3, this summary diagram was constructed to indicate the spatial extent and relative magnitude of the beam distortion that would be generated in the bare and insulated beam tubes as a result of the gas temperature fluctuations. The beneficial effects of reduction in the magnitude and extent of beam distortion by insulating the beam tubes are clearly evident.

It must be reemphasized that the results inferred by Figures 2 through 4 were obtained with a pure diffusion model for the transport of heat between the beam tube wall and internal beam tube gas. In all actuality, thermal gradients generated within the beam tubes will induce convective flow cells that will enhance the transport and distribution of the heat throughout the beam tube gas. The Rayleigh number, Ra , is frequently used as a measure of the strength of a free convection flow. This dimensionless number is given as [2]

$$Ra_L = \frac{g[2 / (T_s + T_{env})](T_s - T_{env}) L^3}{\nu \alpha}, \quad (8)$$

Note: The darker the shading, the more intense the local beam distortion.

Figure 4. Schematic diagram of the relative magnitude and spatial extent of beam distortion in an (a) uninsulated and (b) insulated quad beam tube exposed to an oscillating outer air temperature.

where L is a characteristic length.

For the uninsulated beam tube analysis used to generate the data in Figure 2, calculations predict a Raleigh number on the order of 2×10^6 . Incropera and DeWitt [2] indicate that for a horizontal rectangular enclosure with the bottom surface hotter than the top, instabilities will lead to free convection flows for Ra_L greater than 1,708. Consequently, complex convection flows should develop in the beam tubes. These flows will aid in distributing the heat and thus will prevent sharp temperature gradients from developing near the beam tube walls as shown previously in Figure 2. The net result of such an effect would be a temperature profile that is more closely linear rather than possessing sharp gradients near the tube walls and a flat inner region such as that predicted with the heat diffusion model discussed earlier. This idea is summarized and conveyed in Figure 5. The next section of this report assesses the beam pointing error that would result from a linear temperature gradient across the beam tube.

2.3 Linear Beam Tube Gas Temperature Profile Beam Pointing Error Analysis

Assuming that the temperature profile between two opposing beam tube surfaces is fairly linear, Equation (1) can be used to predict the net beam pointing error. In this section, a simplistic model was constructed to predict this beam pointing error for a variety of different environmental conditions. In this model, it was assumed that the maximum temperature excursion between two opposing beam tube surfaces (as discussed in Section 2.1 of this report) corresponds to the worst-case *linear* temperature profile across the beam tube, as far as a beam pointing error is concerned.

The beam pointing error predicted by Equation (1) requires knowledge of the beam path length, L , and width, S . A path length of 50.0 m and a beam width of 1.25 m were used to model the beam tube geometry of the NIF switchyard and target bay.

This procedure was repeated for several different air temperature amplitudes and periods, as well as insulation thicknesses. For the insulation cases, h was replaced in Equation (4) with an overall heat transfer coefficient, U , that accounted for the insulation thickness, L_{in} , and insulation thermal conductivity, k_{in} ,

$$U = [L_{in}/k_{in} + 1/h]^{-1}. \quad (9)$$

Tuff-R insulation by the Celotex Corporation was used in the present model. Tuff-R is a foam board insulation with aluminum foil coatings on both faces and a thermal conductivity of 0.01 W/m² K.

The maximum temperature differences between the beam tube surfaces and the corresponding beam pointing angle, given by Equation (1), are displayed in Table 1. For the foreseeable HVAC operating conditions given in Table 1, the NIF laser beams can expect to have a total maximum beam pointing error in the range of 0.018 μ rad to 1.025 μ rad resulting from the fluctuating environmental air temperature.

Figure 5. Temperature profiles in the beam tube gas at a particular instant in time as predicted by various thermal models.

Table 1. Summary of the HVAC operating conditions and associated worst-case beam pointing errors.

HVAC Temperature Amplitude (°C)	HVAC Temperature Period (min)	Insulation Thickness (mm)	Maximum Beam Tube Temp. Gradient ΔT (°C)	Maximum Beam Pointing Error $\Delta\theta$ (μ rad)
0.08	10	0	0.0029	0.103
0.08	15	0	0.0043	0.153
0.18	15	0	0.0095	0.338
0.28	15	0	0.0150	0.534
0.38	15	0	0.0200	0.712
0.28	30	0.0	0.0288	1.025
0.28	15	6.4	0.0038	0.135
0.08	15	12.8	0.0005	0.018
0.28	15	12.8	0.0019	0.068

It must be emphasized that the beam pointing errors correspond to a worst-case scenario in which the following conditions must hold:

- A convection heat transfer coefficient differential of $4 \text{ W/m}^2 \text{ K}$ must exist across the beam tube for its entire length from the beam injection to the target bay.
- The temperature excursions of the HVAC systems for the switchyard and target bay in the NIF building must be synchronized.
- The gas within the beam tube must develop a linear temperature gradient during the temperature oscillations between the two opposite beam tube surfaces.
- The beam shot time must occur at the same instant that the opposite beam tube surfaces are reaching their maximum temperature excursion.
- Thermal communication between the four sides of the beam tube must be negligible.

The addition of Tuff-R insulation to the beam tube surfaces results in a significant decrease in the maximum temperature difference between opposing beam tube surfaces and a corresponding decrease in the beam pointing error. Adding 12.8 mm thick insulation will reduce the beam pointing error by a factor of nearly 8.

2.4 Conclusions

Operating limitations in the NIF HVAC system will lead to periodic oscillation in environmental air temperature. The fluctuation in the air temperature will cause a corresponding oscillation in the wall temperatures of the beam transport tubes which will result in thermal fluctuations and gradients in the internal beam tube gas. The net effect will be a distortion of the laser beams as they propagate through the nonuniform gas. In this study, two scenarios were considered to attempt to assess the amount of beam distortion that would exist for anticipated HVAC operating conditions. In both analyses, the results indicated that significant amounts of beam distortion could develop. However, the analyses further dictated that the beam distortion can be decreased by

- reducing the HVAC air temperature amplitude excursion
- reducing the period of the oscillating HVAC air temperature excursion
- adding insulation to the outer beam tube surfaces
- increasing the insulation thickness.

The addition of insulation to the beam tube walls is the best preventative measure against beam distortion. Insulation will greatly reduce the amount of beam distortion,

thus allowing delivery of a greater amount of energy to the laser target. In addition, the presence of insulation serves as a safety cushion against unforeseeable changes or disturbances to the control of the environmental air temperature within the NIF.

3. Light Fixture Evaluation

The fluorescent light fixtures present in the NIF building will serve as heat loads to the HVAC system and may potentially introduce beam pointing errors by causing nonuniform temperature distributions on the laser beam tubes. This section of this report describes the heat generation and dissipation from a fluorescent light bulb fixture with the aid of experimental temperature data from an actual fixture. In addition, a numerical and an analytical investigation used to quantify the thermal interaction of a light fixture with a beam tube will be presented. The numerical model investigates the extent of the convective thermal plume from the light fixture while the analytical model studies the thermal radiation interaction between the light and beam tube. Thus, caution must be exercised in relying solely on the separation distances predicted by just one of these models. Rather, the results of both analyses must be used together to accurately and reliably determine allowable light-fixture to beam-tube separation distances to satisfy beam pointing error limits.

3.1 Empirical Light Fixture Temperature Data

Table 2 displays experimental temperature data obtained from a fluorescent light fixture containing two 4 ft long bulbs that each dissipate 34 W. Figure 6 shows the temperature measurement locations on the light fixture assembly including the bulb surface, the metal fixture housing, and the plastic cover. The room air temperature values at the time the light fixture temperature measurements were made are also presented in Table 2. The last column in Table 2 gives light fixture temperatures that have been normalized to an air temperature of 26°C. These normalized temperatures were obtained

Figure 6. Schematic diagram of the temperature measurement locations on a fluorescent light fixture assembly (a) without cover plate and (b) with cover plate.

by adding the difference between 26°C and the true air temperature to the corresponding light fixture temperature measurement.

Table 2. Experimental temperature measurements from various locations on a fluorescent light fixture containing two 4 ft long 34 W bulbs.

Measurement Location	Temperature (°C)	Air Temperature (°C)	Normalized Light Fixture Temperature (°C). [Corresponds to an air temperature of 26°C]
Bulb Surface (T1)	43.6	25.6	44.0
Fixture Housing (T2)	35.9	25.6	36.3
Fixture Housing (T3)	34.7	27.3	33.4
Fixture Housing (T4)	45.2	27.3	43.9
Plastic Cover (T5)	27.1	21.4	31.7
Plastic Cover (T6)	26.6	21.4	31.0
Fixture Housing (T7)	39.1	25.5	39.6
Fixture Housing (T8)	48.8	25.5	49.3

The temperature data of Table 2 provided vital information on the heat dissipation characteristics of a fluorescent light fixture that was required for the analytical and numerical models presented in the next section of this report. In particular, the temperature data were used to determine the modes and routes of energy dissipation from the light fixture components.

The two primary modes of heat transfer from the light fixture components were free convection, q_{conv} , and radiation, q_{rad} . Assuming that the light fixture behaves as a diffuse gray body in a large environment, these heat transfer rates can be expressed as [2]

$$q_{conv} = h A (T_s - T_{env}) \quad (10)$$

and

$$q_{rad} = \varepsilon \sigma A (T_s^4 - T_{env}^4), \quad (11)$$

where T_s and T_{env} are the temperatures of the fixture component and environment, respectively, ε is the hemispherical emissivity of the fixture component, and σ is the Stefan-Boltzmann constant ($5.670 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$).

The heat transfer coefficient for the light bulb was determined using the following empirical Nusselt correlation for a horizontal cylinder with diameter d [2],

$$\bar{h} = \frac{k}{d} \left\{ 0.60 + \frac{0.387 Ra_d^{0.167}}{\left[1 + (0.559 / Pr)^{0.563} \right]^{0.296}} \right\} \quad (12)$$

where the Rayleigh number, Ra , was given previously in Equation (8).

Using appropriate air properties with air and bulb surface temperatures of 25.6°C and 43.6°C, Equations (10) through (12) predicted that the fluorescent bulb used in this study dissipated approximately 38% (13 W) of its energy by convection and 52% (21 W) by radiation. This finding is fairly consistent with the convective and radiative percent values of 29% and 58%, respectively, for a fluorescent bulb as given in [6].

The differences in fixture housing surface temperature measurements T3 and T4 indicate that the ballast is a significant heat source and should be accounted for in an energy balance on the light fixture. The ASHRAE Fundamentals Handbook [7] suggests that typical older fluorescent light ballasts of the type used in this study may dissipate approximately 20% of the power rating of the light bulbs, or 13.6 W. Newer, more

efficient ballasts, like those to be used in the NIF light fixtures, may dissipate closer to 10% of the power rating of the light bulbs.

To determine the proportion of heat dissipated by convection and radiation from the light fixture housing and plastic cover, the following Nusselt correlation was employed to determine the average free convection heat transfer coefficient for the entire light fixture [2]:

$$h_L = 0.27 \frac{k}{L} Ra_L^{0.25}, \quad (13)$$

where L is a characteristic length given as

$$L = \frac{A_s}{P}, \quad (14)$$

where A_s is the surface area of the plate and P is the perimeter.

Equation (13) was derived for a downward facing heated surface and was thus determined appropriate for calculating the heat transfer coefficient for the bottom surface of the light fixture. For an upward facing heat surface, a different empirical Nusselt correlation would be needed. However, the narrow gap (1.5 cm) that existed between the top of the light fixture housing and the ceiling reduced the heat removal efficiency and thus did represent the physical situation to allow a Nusselt correlation for an upward facing plate to be used for the top surface of the light fixture. Consequently, it was

approximated that the heat transfer coefficient predicted from Equation (13) was representative of the entire light fixture housing.

An average temperature difference of 12°C between the light fixture and air gives a Rayleigh number of 1.94×10^6 and an average heat transfer coefficient of 2.2 W/m²K for the light fixture surface using Equation (13). Using this heat transfer coefficient as well as the average surface areas and temperatures of the light fixture metal housing (mh) and plastic cover (pc), Equation (10) predicts that the light fixture housing dissipates roughly 26 W by free convection:

$$q_{conv} = h A_{mh} (T_{mh} - T_{env}) + h A_{pc} (T_{pc} - T_{env}) = 26 \text{ W},$$

where $A_{mh} = 0.507 \text{ m}^2$,

$$A_{pc} = 0.468 \text{ m}^2,$$

$$T_{mh} = 44.5^\circ\text{C},$$

$$T_{pc} = 31.4^\circ\text{C}, \text{ and}$$

$$T_{env} = 26^\circ\text{C}.$$

The remaining 55.6 W of power in the light fixture is dissipated by radiation heat transfer.

Figure 7 summarizes the heat dissipation characteristics of the various light fixture components. Each light bulb generates 34 W and the ballast is assumed to generate 13.6 W for a total of 81.6 W for the light fixture assembly. Figure 7(a) shows

(a)

(b)

Figure 7. Schematic diagram of the heat flow distribution from a fluorescent light fixture assembly (a) without cover plate and (b) with cover plate.

that without a plastic light cover fixture, each light bulb surface dissipates approximately 13 W by free convection, while the remaining 21 W of power is dissipated by radiation heat transfer. Figure 7(b) shows the heat dissipation characteristics of the light fixture with the plastic cover in place. The plastic cover dissipates 5.6 W by convection while the hotter metal fixture housing dissipates 20.4 W. The remaining 55.6 W is transferred in the form of radiation from all of the light fixture components to the environment. The majority of this radiation is emitted from the light bulbs and passes through the plastic cover to the environment beneath the light fixture. The remainder of the radiation is emitted diffusely from the hot light fixture housing and plastic cover.

3.2 CFX Model of Light Fixture Thermal Plume

Uniform and bounded air temperature fields are required within the vicinity of the NIF beam tubes to prevent unwanted beam pointing errors resulting from temperature gradients. However, the light fixtures within the NIF switchyard and target bay act as localized heat sources and thus present a potential for relatively large thermal gradients within the HVAC system airflow. Consequently, a safe working distance between the light fixtures and the beam tubes is required to prevent undesired performance. In order to establish design criteria for the light-fixture to beam-tube spacing, an accurate thermal model was needed to predict the extent and distribution of the thermal plume from the light fixture into the airflow, as conveyed in Figure 8.

The numerical analysis of the thermal plume from the light fixture into the airflow involves simultaneously solving the continuity, Navier-Stokes (momentum), and energy equations. This was achieved using the computational fluid dynamics software package CFX [8]. CFX is a finite volume, implicit Navier-Stokes solver that uses a revised version of the mathematical algorithm Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) [9] termed SIMPLEC [10]. In addition, the code uses several well-developed differencing schemes to discretize the derivatives in the governing differential equations [8]. Turbulence created in the flow was predicted with a two-equation K - ϵ model using CFX default values for K and ϵ . CFX is an industry-driven code that has been developed under ISO 9001 requirements and has been validated with numerous test problems.

Figure 8. Schematic of the thermal plume development in the HVAC airflow caused by the heat supply of the light fixture.

3.2.1 Model Geometry, Boundary Conditions, and Grid Discretization

The geometrical model for the numerical simulations is shown in Figure 9. A two-dimensional geometry was used to simplify the problem but still yield meaningful, worst-case results. The light fixture was modeled as a constant temperature surface, with the temperature selected to supply a representative convective heat flow into the HVAC airflow. Table 3 lists the types of fluorescent light fixtures being modeled, the amounts of heat dissipated by convection and radiation heat transfer from each fixture, and the corresponding surface temperature required to dissipate the convective heat flow. The percentages of heat dissipated by convection (32%) and radiation (68%) were determined from the experimental light fixture study presented earlier (see Figure 7). Radiation heat transfer was not modeled in this study because the feature of interest was the convective thermal plume from the light fixture. Since air is accurately modeled as a non participating radiation medium, radiation heat transfer would not influence the thermal plume in this study.

Table 3. Temperature and heat transfer rate dissipation characteristics of fluorescent light fixtures used in the CFX models.

Light Fixture	Total Power Dissipation (W) (10% Ballast)	Convection Heat Transfer (W) (32% of Total)	Radiation Heat Transfer (W) (68% of Total)	CFX Model Light Surface Temperature (K)
Three, 32 W fluorescent bulbs	105.6	33.8	71.8	330
Two, 32 W fluorescent bulbs	70.4	22.5	47.9	318

To represent the downward airflow from the vented ceiling ducts, a uniform inlet airflow profile was established on either side of the light fixture. Air at atmospheric pressure and 20.0°C (293.0 K) was used as the cooling medium with inlet velocities of 0.015 m/s (3 fpm) and 0.063 m/s (12.5 fpm). The velocities are representative of the inlet air velocities being considered for the laser bay and switchyard HVAC systems.

The width of the model was based upon the 3.048 m (10 ft) center-to-center spacing of the majority of the light fixtures in the NIF switchyard as specified in the NIF Beam Transport System Title II design drawings. Thus symmetry boundaries were placed on either side of the light fixture model. The extent of the model in the flow direction was made sufficiently long to prevent any downstream boundary conditions from communicating information back upstream. In this model, the beam tube was not included since the thermal plume was the feature of interest. A more detailed thermal model, which contains the beam tube, is currently being developed.

Other modeling conditions included steady-state buoyant and turbulent flow, as well as convection heat transfer. In addition, since the temperature and density variations were relatively small, the Boussinesq approximation could be made which allowed the air to be modeled as an incompressible medium.

The grid generator for CFX uses a multi-block scheme with a body-fitted grid structure. In this study, a uniform grid with a spacing of 2.54 cm (1 in.) was used throughout the model.

Note: All Dimensions in Meters

Figure 9. Geometrical model and boundary conditions for the CFX light fixture analysis.

3.2.2 Convergence Criteria

To determine the convergence and credibility of the CFX model, a three-step validation procedure of the numerical solution was employed. First and foremost, the convergence of a model onto a solution was determined with *residuals*. In the solution algorithm of CFX, each dependent variable, y (i.e., temperature), is solved for in algebraic equations of the form

$$f(y) = F \quad (15)$$

where F is a forcing function.

The residual for the dependent variable, y , is the error or difference between the left and right hand sides of Equation (15). The residuals for each dependent variable were determined at each cell and summed over all cells for a single iteration. A converged solution was obtained when the sum of the residuals for each dependent variable ceased to vary between consecutive iterations.

The final two tests of convergence were mass and energy balances of the model system. For the steady-state problems considered in this study, convergence was assumed when there was less than a 5% difference in the inlet and outlet flows of mass and energy.

3.2.3 Results

The numerical predictions of the velocity and temperature fields downstream of the fluorescent light fixture are presented in Figures 10 through 14 for several different operating conditions. The parameters that varied in this study include

- inlet air velocity,
- heat dissipation rate of the light fixture (number of light bulbs), and
- center-to-center light fixture spacing.

The current NIF switchyard design calls for fluorescent light fixtures with three, 32 W bulbs on a 3.05 m (10 ft) center-to-center spacing (perpendicular to largest length of fixture) and an inlet HVAC air velocity of 0.064 m/s (12.5 fpm) at a mean temperature of 293.00 K. The numerical velocity vector plot predicted by CFX for this situation is shown in Figure 10. The vector plot indicates that the airflow, after entering through the inlet, is accelerated and pushed outward as it passes by the light fixture. This flow diversion is caused by the development of the hot thermal plume of air that is heated and forced horizontally outward away from the light surface. A small recirculating plume is developed beneath the light fixture to direct cool air upward to replace the heated air. The corresponding temperature field is displayed in Figure 11 over three different temperature scales. The temperature contours reveal the development of a hot thermal plume which develops beneath the light fixture and spreads out in an umbrella-like shape. The hot thermal plume is dissociated by the incoming cooled air through diffusion and

Figure 10. Numerical model velocity vector plot of the light fixture thermal plume for a light surface temperature of 330 K, a total convective heat dissipation rate of 33.4 W, and an inlet air velocity of 0.064 m/s.

Figure 11. Numerical model temperature contour plots of the light fixture thermal plume at different temperature scales for a light surface temperature of 330 K, a total convective heat dissipation rate of 33.4 W, and an inlet air velocity of 0.064 m/s, showing beam tube positioning for steady-state beam pointing errors of (a) 10.0 μ rad, (b) 1.0 μ rad, and (c) 0.1 μ rad.

advection of energy. Far downstream of the light fixture, the air temperature becomes nearly uniform as the flow becomes fairly well mixed.

The thermal plume is seen to extend a significant distance away from the light fixture. In fact, as the different temperature scale contour plots of Figure 11 indicate, the extent and strength of the thermal plume depends upon the scale or decimal precision used to track the temperature of the heated air. Consequently, no exact safe operating distance of the beam tube from the light fixture can be stated without specifying the acceptable operating conditions of the beam tube. In particular, the required separation distance will depend on the maximum acceptable steady-state beam pointing error and the maximum allowable air temperature that the beam tube may be exposed to.

If a beam tube is placed within the steady-state thermal plume, distortion of the beam will result from the establishment of a nonuniform temperature field within the beam tube. If it is assumed that a linear temperature profile is established from the top to the bottom of the beam tube, then a steady-state beam pointing error [given earlier by Equation (1)] will be developed. By selecting different beam pointing errors and determining the associated temperature differentials from Equation (1) (for a beam tube length and width of 50 m and 1.2 m, respectively), the approximate location of the beam tube beneath the light fixture can be determined. This finding is demonstrated in Figure 11, where three different beam tube locations have been determined for three different acceptable steady-state beam pointing errors of 10.0, 1.0, and 0.1 μrad . These beam pointing errors encompass the realistic range that could be corrected with the beam alignment system. In each case, the maximum air temperature that the beam tube is exposed to is also indicated. As expected, as the beam pointing error requirement is

relaxed from 0.1 to 10.0 μrad , the minimum light-fixture to beam-tube spacing is decreased.

This beam tube locating technique was repeated for several different operating scenarios including a 33% reduction in the light fixture heat dissipation, a 75% reduction in airflow rate (in addition to the 33% reduction in power), and 200% increase in the separation distance between light fixtures (in addition to the previous two reductions). The numerical velocity vector and temperature contour plots for these three cases are displayed in Figures 12 through 14.

Figure 12. Numerical model (a) velocity vector plot and (b) temperature contour plot of the light fixture thermal plume for a light surface temperature of 318 K, a total convective heat dissipation rate of 21.2 W, and an inlet air velocity of 0.064 m/s.

Figure 13. Numerical model (a) velocity vector plot and (b) temperature contour plot of the light fixture thermal plume for a light surface temperature of 318 K, a total convective heat dissipation rate of 21.2 W, and an inlet air velocity of 0.015 m/s.

Figure 14. Numerical model (a) velocity vector plot and (b) temperature contour plot of the light fixture thermal plume for a light surface temperature of 318 K, a total convective heat dissipation rate of 21.2 W, an inlet air velocity of 0.015 m/s and a center-to-center light spacing of 6.1 m (20 ft).

Table 4 summarizes the minimum light-fixture to beam-tube spacing for all of the models developed in this study, based on an allowable beam pointing error of 0.1, 1.0, and 10.0 μrad (and the associated temperature drop across the beam tube from top to bottom) and a maximum air temperature that the beam tube is exposed to.

The data in Table 4 indicates that the minimum light-fixture to beam-tube spacing can be reduced by

- reducing the number of 32 W bulbs in the light fixture from three to two,
- increasing the inlet air velocity,
- increasing the center-to-center distance between the light fixtures, and
- increasing the allowable steady-state beam pointing error.

Table 4. Minimum light-fixture to beam-tube spacing for various operating conditions as predicted by the CFX model of the light fixture thermal plume with an inlet air temperature of 293.00 K.

Inlet Air Velocity (m/s)	Number of 32 W Bulbs in Light Fixture	Center-to-Center Light Fixture Spacing (m)	Allowable Steady-State Beam Pointing Error, $\Delta\theta$ (μrad)	Allowable ΔT_{max} (K) Corresponding to Beam Pointing Error	Maximum Air Temperature Seen by Beam Tube (K)	Minimum Light-Fixture to Beam-Tube Separation Distance (m) [ft]
0.064	3	3.05 [10 ft]	0.1	0.007	293.15	6.01 [19.72]
0.064	3	3.05	1.0	0.070	293.21	2.80 [9.19]
0.064	3	3.05	10.0	0.700	293.28	1.65 [5.41]
0.064	2	3.05	0.1	0.007	293.08	5.94 [19.48]
0.064	2	3.05	1.0	0.070	293.18	2.80 [9.19]
0.064	2	3.05	10.0	0.700	293.28	1.53 [5.02]
0.015	2	3.05	0.1	0.007	293.40	2.44 [8.01]
0.015	2	3.05	1.0	0.070	293.40	0.76 [2.49]
0.015	2	0.305	10.0	0.700	293.40	0.69 [2.26]
0.015	2	0.610 [20 ft]	0.1	0.007	293.14	2.74 [8.99]
0.015	2	0.610	1.0	0.070	293.14	1.22 [4.00]
0.015	2	0.610	10.0	0.700	293.14	0.91 [2.99]

The shaded boxes in Table 4 are used to highlight acceptable light-fixture to beam-tube separation distances (less than 3.05 m) and a maximum beam tube temperature of 293.28 K (for an air temperature of 293.00 K). Several different operating scenarios will produce allowable beam pointing errors as predicted with the CFX model of the thermal plume. *Caution must be exercised in relying solely on the separation distances of Table 4 for design purposes as the CFX model used to generate them did not account for radiation interaction between the light fixture and beam tube.* The radiation interaction is investigated with an analytical model in the following section of this report. The assessment of the next section, combined with the data of Table 4, will allow accurate and reliable separation distances to be selected.

3.3 Analytical Thermal Model of Light Fixture and Beam Tube

To predict the radiation thermal interaction between a fluorescent light fixture and a laser beam tube, a lumped capacitance analytical model was developed. While this model is not capable of predicting the complex features of the thermal plume from the light fixture (as was performed with the CFX model in the previous section), it does provide valuable information concerning the exchange of radiation heat transfer between the light fixture and beam tube. The beam tube configuration and light fixture layout in the NIF switchyard design are quite complex. Thus, the simplified geometry shown in Figure 15 was considered that represented the most general geometrical relation between the light fixture and beam tube. The majority of the lighting provided in the current NIF switchyard design is provided by fluorescent light fixtures, each with three, 32 W fluorescent bulbs in a 1.22 m by 0.61 m (4 ft by 2 ft) housing. Figure 15 shows how a single light fixture and a portion of a beam tube were selected for the analytical model. Symmetry planes from neighboring light fixtures allowed such geometry to be selected.

Based on the geometry of Figure 15, the analytical thermal radiation resistance model depicted in Figure 16 was generated. The thermal resistance network describes the heat transfer processes between the light fixture, beam tube, and environment, based upon the boundary conditions, geometrical parameters, as well as surface and fluid properties. A more complete discussion of thermal radiation resistance models can be found in [2]. The radiation heat transfer rate boundary condition of 69 W for the fluorescent light fixture was taken as 60% of the 115 W dissipated by the light bulbs and ballast. This percentage is consistent with the experimental light fixture thermal assessment presented earlier. An external heat transfer coefficient of 1.0 W/m² K was selected for the

convection heat transfer from the beam tube surface. This value was selected based upon the HVAC analysis presented earlier [Equation (5)] and represents a worst-case scenario. By selecting the lowest possible (yet realistic) heat transfer coefficient, the means to convectively cool the beam tube surface is minimized and the light-fixture to beam-tube separation distance is maximized. In addition, the beam tube and environmental air temperatures were chosen as 293.28 K and 293.00 K, respectively, to represent a worst-case scenario. It is assumed that the beam tube temperature of 293.28 K would correspond to the beam tube surface exposed directly to the light, while the 293.00 K air temperature would correspond closely to the temperature on the opposing side beam tube surface. This worst-case temperature difference, if producing a linear, steady-state temperature profile across the beam tube, would lead to a steady-state beam pointing error of $10.0\ \mu\text{rad}$ for the switchyard and target bay beam tubes with an approximate length and width of 50 m and 1.25 m, respectively. Since this is a steady-state beam pointing error, it may be corrected with the beam alignment system. However, this steady-state beam pointing error serves as an upper limit to aid in determining the location of the beam tube in reference to the light fixture. *Consequently, the purpose of the analysis was to determine the minimum allowable separation distance between the light fixture and beam tube to satisfy the model boundary conditions which in turn match the steady-state beam pointing error limitation of $10\ \mu\text{rad}$.*

Figure 15. Schematic diagram of the light fixture/beam tube geometry used in the analytical radiation heat transfer model.

Figure 16. (a) Schematic diagram of the light fixture/beam tube and (b) the corresponding radiation resistance network model.

In addition to the boundary conditions, the following assumptions were made:

- steady-state heat transfer rates,
- diffuse, gray surfaces,
- all radiation heat dissipation by the light source is directed downward, or ± 90 degrees from downward pointing normal of the light fixture,
- air is a non radiative participating medium,
- uniform surface temperatures across beam tube and light fixture,
- no interaction between the thermal plume of the light fixture and the beam tube exists (The beam tube is located outside of this region which was assessed in the previous section of this report with the CFX model), and
- environment is represented as a hypothetical surface which is approximated as a black body at 293.0 K

The analytical analysis required satisfying energy balances on the light fixture and beam tube while using the radiation resistance network of Figure 16. This analysis, which is outlined in Appendix A, generates the following three equations which must be solved simultaneously for the view factor, F_{12} , between the beam tube and light fixture:

$$J_1 = h A(T_1 - T_3) \left[\frac{1 - \epsilon_1}{\epsilon_1 A_1} \right] + \sigma T_1^4, \quad (16)$$

$$J_2 = \frac{1}{F_{12}^{-1}} \left\{ h A \frac{(T_1 - T_3)}{\varepsilon_1 A_1} + \sigma (T_1^4 - T_3^4) + \sigma T_3^4 F_{12} \right\}, \quad (17)$$

and

$$q_{rad} = J_2 A_2 - J_1 A_1 F_{12} - A_2 \sigma T_3^4 \left(1 - \frac{A_1}{A_2} F_{12} \right), \quad (18)$$

where J_1 and J_2 are the surface radiosities from the beam tube and light fixture, respectively.

Once F_{12} is solved for from Equations (16) through (18), the following view factor relation for the light fixture and beam tube (parallel plates with midlines connected by a perpendicular) can be used implicitly to back out the separation distance, L , between them:

$$F_{12} = \frac{A_2}{A_1} \frac{\left[(W_2 + W_1)^2 + 4 \right]^{0.5} - \left[(W_1 - W_2)^2 + 4 \right]^{0.5}}{2 W_2}, \quad (19)$$

where

$$W_1 = \frac{w_1}{L}, \quad W_2 = \frac{w_2}{L},$$

and w_1 and w_2 are the beam tube width and light fixture width, respectively. The view factor relation given by Equation (19) assumes that the light fixture is positioned directly above the beam tube. If the beam tube is not located directly beneath the light fixture, its view factor and corresponding operating temperature will be less than those for a beam tube located directly beneath the light with the same mean separation distance.

Consequently, the minimum required separation distance predicted by this analysis will correspond to a worst-case scenario.

This analysis was conducted for bare (steel) and insulated (foam insulation with a shiny aluminum foil outer layer) beam tubes with hemispherical emissivities of 0.30 and 0.07, respectively. In addition, studies were performed for the current fluorescent light fixture design with three 32 W bulbs and for an alternative light fixture that houses two 32 W bulbs. The required separation distances between the light fixture and beam tube for these cases are listed in Table 5.

The analysis indicates that the noninsulated beam tube surface will absorb a relatively high amount of radiation from the light fixtures, requiring a separation distance of at least 5.7 m (18.7 ft) to satisfy the operating temperature condition for the large three bulb light fixture. By reducing the light fixture power by 33%, the required separation distance is reduced to 3.5 m (11.5 ft).

The addition of a radiation shield such as aluminum foil to the outside of the beam tube will significantly reduce the radiation heating from the light fixtures and require a minimum separation distance of only 2.5 m (8.2 ft) for the three bulb fluorescent light fixture. By reducing the light fixture power by 33% with the addition of the radiation shield, the required separation distance is reduced to 1.6 m (5.2 ft). These later two separation distances with the radiation shield in place are more reasonable and are consistent with the separation distances required to avoid thermal plume interaction from the light fixtures, as found previously with the CFX model.

Table 5. Minimum light-fixture to beam-tube separation distances as predicted by the analytical radiation analysis.

Number of 32 W bulbs in Light Fixture	Beam Tube Surface	Minimum Light-Fixture to Beam- Tube Separation Distance (m) [ft]
3	No insulation, stainless steel outer surface with $\varepsilon = 0.30$	5.7 [18.7]
2	No insulation, stainless steel outer surface with $\varepsilon = 0.30$	3.5 [11.5]
3	Insulation, aluminum foil with $\varepsilon =$ 0.07	2.5 [8.2]
2	Insulation, aluminum foil with $\varepsilon =$ 0.07	1.6 [5.2]

The minimum light-fixture to beam-tube separation distances given by the analytical model in Table 5 are larger than those determined by the CFX thermal plume analysis in the previous section. Thus it appears that radiation heat exchange between the light fixture and beam tube, rather than the presence of the thermal plume, will dictate the positioning of the beam tubes relative to the light fixtures.

3.4 Conclusions

This section presented several different approaches to determine the heat transfer characteristics of a fluorescent light fixture and the interactions of these characteristics with a beam tube. From these studies, the following conclusions can be drawn:

- The power dissipation of a typical fluorescent light fixture is the sum of the power rating of the bulbs plus an additional 10% to 20% to account for thermal losses of the voltage regulator or ballast.
- Of the total power generated by a fluorescent light fixture, roughly 32% is dissipated by convection heat transfer and the remaining 68% by radiation.
- A thermal plume created by the downward movement of the HVAC airflow will extend a significant distance below each light fixture. The temperature gradients within the thermal plume will generate steady-state beam pointing errors if a beam tube is located in the region of the plume. The absolute value of the beam pointing error varies with beam tube location but may be reduced by reducing the value of the light fixture heat dissipation, increasing the HVAC airflow rate and increasing the center-to-center light fixture spacing.
- While the thermal plume analysis revealed the convective heat transfer influence on the beam pointing error, an analytical study of the thermal radiation exchange between the light fixture and beam tube revealed that radiation heat transfer will dictate the minimum light-fixture-to-beam tube separation distance required to satisfy beam pointing error limitations. Using the thermal radiation analysis as a design guide, the minimum light-fixture to beam-tube separation distances were determined

for two different light fixture sizes and beam tubes with and without thermal insulation. These minimum separation distances, determined for a steady-state beam pointing error of $10\ \mu\text{rad}$, are displayed in Table 6.

Table 6. Minimum light-fixture to beam-tube separation distance for a steady-state beam pointing error of $10\ \mu\text{rad}$ as predicted with the thermal light fixture analyses.

Number of 32 W bulbs in Light Fixture	Beam Tube Surface Insulation	Minimum Light-Fixture to Beam-Tube Separation Distance (m) [ft]
3	None	5.7 [18.7]
2	None	3.5 [11.5]
3	1.52 cm (0.5 in.) foam with shiny aluminum foil coating on outside	2.5 [8.2]
2	1.52 cm (0.5 in.) foam with shiny aluminum foil coating on outside	1.6 [5.2]

4. General Conclusions and Design Recommendations

This study investigated the influence of the National Ignition Facility's HVAC and lighting systems on the performance of the NIF beam transport system. Analytical, experimental, and numerical investigations were used to assess the degradation of the NIF laser beams from thermal gradients established inside the beam transport tubes by external thermal sources. The key conclusions from these investigations are as follows:

- The periodic oscillation of the cooling air temperature from the HVAC system was found to lead to transient beam distortion and steering effects that cannot be accounted for or corrected with the present steady-state beam alignment system. The beam distortion can be decreased by reducing the amplitude and period of the oscillating HVAC air temperature and by adding thermal insulation to the outside of the beam tube surfaces. The magnitudes of the beam pointing errors associated with various operations are repeated again in Table 7.

Table 7. Summary of the HVAC operating conditions and associated worst-case beam pointing errors.

HVAC Temperature Amplitude (°C)	HVAC Temperature Period (min)	Insulation Thickness (mm)	Maximum Beam Tube Temp. Gradient ΔT (°C)	Maximum Beam Pointing Error $\Delta\theta$ (μ rad)
0.08	10	0	0.0029	0.103
0.08	15	0	0.0043	0.153
0.18	15	0	0.0095	0.338
0.28	15	0	0.0150	0.534
0.38	15	0	0.0200	0.712
0.28	30	0.0	0.0288	1.025
0.28	15	6.4	0.0038	0.135
0.08	15	12.8	0.0005	0.018
0.28	15	12.8	0.0019	0.068

- The light fixtures were found to generate large thermal plumes in the cooling air as well as serve as a source of significant thermal radiation, both of which would cause significant steady-state beam pointing errors. Unlike the transient effects induced by the HVAC system, the steady-state beam pointing errors imposed by the light fixtures may be corrected by the beam alignment system. The minimum light-fixture to beam-tube separation distances associated with various operations are repeated again in Table 8. These distances were determined for a beam tube situated directly beneath a light fixture. However, as a measure of safety, the separation distance should be interpreted as a separation radius between the center of the light fixture and the outer edge of a beam tube located anywhere beneath the light fixture. For beam tubes located above a light fixture (i.e., not directly illuminated by the light), separation distances of half the values in Table 8 should be employed.

Table 8. Minimum light-fixture to beam-tube separation distance for a steady-state beam pointing error of $10\ \mu\text{rad}$ as predicted with the thermal light fixture analyses.

Number of 32 W bulbs in Light Fixture	Beam Tube Surface Insulation	Minimum Light-Fixture to Beam-Tube Separation Distance (m) [ft]
3	None	5.7 [18.7]
2	None	3.5 [11.5]
3	1.52 cm (0.5 in.) foam with shiny aluminum foil coating on outside	2.5 [8.2]
2	1.52 cm (0.5 in.) foam with shiny aluminum foil coating on outside	1.6 [5.2]

The general design recommendations for NIF, based upon the conclusions of this study, are as follows:

- Reduce the fluorescent light fixture power in the NIF switchyards and target bay by 33% by replacing the 96 W fixtures (three 32 W bulbs) with 64 W fixtures (two 32 W bulbs).
- Insulate the outside of the beam tubes in the NIF switchyards and target bay with 1.27 cm (0.5 in.) thick foam board insulation. The foam insulation should have a thermal conductivity less than or equal to 0.019 W/m K (R value ≥ 4.1) and a shiny aluminum foil outer coating (emissivity ≈ 0.07). Insulation is highly recommended because it will greatly reduce the amount of beam distortion and steering, thus allowing a greater amount of energy to be delivered to the laser target. The insulation will also serve as a safety cushion for unforeseeable changes or disturbances in the NIF environmental control systems.
- Ensure that the minimum light-fixture to beam-tube spacing given in Table 8 is adhered to based upon the final light fixture and beam tube insulation conditions.

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Appendix A

Calculations used in the Thermal Analysis of Light Fixture and Beam Tube

The energy balance on the beam tube can be expressed as

$$q_{conv} = -h A(T_1 - T_3) = \frac{\sigma T_1^4 - J_1}{\frac{1 - \epsilon_1}{\epsilon_1 A_1}}, \quad (\text{A.1})$$

or upon rearranging,

$$J_1 = h A(T_1 - T_3) \left[\frac{1 - \epsilon_1}{\epsilon_1 A_1} \right] + \sigma T_1^4. \quad (\text{A.2})$$

Another form of the energy balance on the beam tube based on net radiation transport is given by

$$\frac{\sigma T_1^4 - J_1}{\frac{1 - \epsilon_1}{\epsilon_1 A_1}} = \frac{J_1 - J_2}{(A_1 F_{12})^{-1}} + \frac{J_1 - \sigma T_3^4}{(A_1 F_{13})^{-1}}. \quad (\text{A.3})$$

Using the following view factor relation

$$F_{12} + F_{13} = 1 \quad (\text{A.4})$$

and substitution of Equation (A.2), Equation (A.3) can be rearranged and manipulated to give,

$$J_2 = \frac{1}{F_{12}^{-1}} \left\{ h A \frac{(T_1 - T_3)}{\epsilon_1 A_1} + \sigma (T_1^4 - T_3^4) + \sigma T_3^4 F_{12} \right\}. \quad (\text{A.5})$$

A similar energy balance of the light fixture yields,

$$q_{rad} = \frac{J_2 - J_1}{(A_2 F_{21})^{-1}} + \frac{J_2 - \sigma T_3^4}{(A_2 F_{23})^{-1}}. \quad (A.6)$$

Using the view factor relations

$$A_2 F_{21} + A_2 F_{23} = 1 \quad (A.7)$$

and

$$A_2 F_{21} = A_1 F_{12}, \quad (A.8)$$

Equation (A.6) can be rearranged to give

$$q_{rad} = J_2 A_2 - J_1 A_1 F_{12} - A_2 \sigma T_3^4 \left(1 - \frac{A_1}{A_2} F_{12} \right). \quad (A.9)$$

The view factor relationship for parallel plates with midlines connected by a perpendicular is given by [2]

$$F_{12} = \frac{A_2}{A_1} \frac{\left[(W_2 + W_1)^2 + 4 \right]^{0.5} - \left[(W_1 - W_2)^2 + 4 \right]^{0.5}}{2 W_2}, \quad (A.10)$$

where

$$W_1 = \frac{w_1}{L}, W_2 = \frac{w_2}{L},$$

and w_2 , w_1 , and L are the light width, beam tube width, and separation distance, respectively.