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Mechanical and optical analysis of large-aperture optics mounted on a frame with a curved surface

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ABSTRACT

Motivated by the need to decrease gravitational sag and stress and increase the frequency conversion efficiency of large-aperture optics, a novel mounting configuration in which a set of optics is mechanically mounted to a frame with a curved surface was developed and mechanically and optically analyzed. The effects of an external load on distortion, stress and the induced frequency conversion efficiency were studied, and the changes in these values with varying external load were assessed to determine the optimum mounting configuration. Additionally, the effects of the frame surface topography on distortion, stress and the induced frequency conversion efficiency were studied. The optimum values for distortion, stress and the induced frequency conversion efficiency were determined for different topographies.

1. Introduction

With the use of frequency conversion, the applications of nonlinear optics have expanded and now include inertial confinement fusion [1]. Due to their excellent frequency conversion performance [2], KDP crystals are used as frequency conversion optics [3] for Inertial Confinement Fusion (ICF) devices, such as NIF [4, 5] and Omega [6] in the USA, LMJ [7] in France, and SG-II [8] and SG-III [9] in China. The optics in these ICF devices generally have large apertures and thin thicknesses, resulting in gravitational sag due to their own weight, thus introducing distortion, stress and changes in the incident angle and refractive index. These changes cause phase mismatches and ultimately reduce the frequency conversion efficiency [10–12].

The key to solving this problem is appropriately mounting the optics to minimize gravitational sag. A variety of techniques have been proposed for use in ICF. P. J. Wegner et al. [13] supported the optics at the corners with compliant polymer spacers and used stainless steel buttons to press against the optics at the compliant spacers. C. E. Barker et al. [14] proposed three types of mounting configurations for simple supports at all four corners and edges, coupled with clamping supports at all four edges. Additionally, the gravitationally induced distortion was calculated using a finite element method. J. M. Auerbach et al. [15] extended C. E. Barker's models to calculate the detuning angle distributions, which related the optical performance to the distortion. R. L. Hibbard et al. [16] designed a mechanical configuration in which the optics are mounted against the frame surface and the other side is a compliant element that applies an external load to the optics through mechanical deformation. Olivier Lubin et al. [17] bonded the optics to the mounting frame at the vertical edges with a silicone seal.

The mounting configurations for large-aperture optics in other optical fields are suggestive and could be applied in ICF. A. Nordt et al. [18] axially fixed the optics to a frame with bolts and provided a spring-loaded radial preload. E. T. Kvamme et al. [19] axially pushed the optics against the frame using springs and spacers and radially loaded the edge using a flexure ring and spacers. G. Kroes et al. [20] axially supported the optics by directly pushing them against the frame surfaces and radially supported them at the edges. Anlu Li et al. [21,22] proposed three types of mounting configurations, point-contact, full-contact and partial surface-contact supports, in which the optics are axially and radially supported with a flexible material. Lijuan Wang et al. [23] used steel straps and nylon pads to support the optics axially and radially, whereas Anlu Li et al. [24] analyzed the distortion and stress in this configuration with a finite element method. B. Saggin et al. [25] radially bonded the optics at the edges with elastic blades that were fixed to the frame. D. M. Stubbs et al. [26] mounted the optics in a semi-kinematic six-strut flexure arrangement and radially bonded it at the edges. C. L. Hom et al. [27] bonded the optics to the frame with pads and axially loaded them with flexible monopods to minimize bending.

As a result of fabrication, the frame surface has an imperfect topography and a curved surface, which has significant effects on distortion and stress and further affects the frequency conversion

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efficiency because the optics rest directly against the curved surface. Unfortunately, effects of the curved surface were not assessed in the previous work, and the frame surface was assumed to be ideally smooth. In this paper, the effects of a curved frame surface were taken into consideration; a model of a mounting configuration with the optics mounted on a frame with a curved surface was created for mechanical and optical analysis. The effects of external loading on distortion, stress and the induced second harmonic generation (SHG) efficiency were studied. The Root Mean Square (RMS) values of the distortion and stress, as well as the induced SHG efficiency decrease, were calculated. The changes in these variables with varying external load were determined, and the optimal values were identified. Moreover, the effects of the frame surface topography on distortion, stress and the induced SHG efficiency were studied by examining a series of surface flatnesses to represent the topography. The optimal RMS values for distortion, stress and the induced SHG efficiency decrease for each flatness were calculated, and their changes with varying surface curvature were determined.

2. Optical and mechanical mounting configurations

2.1. Optical configuration of the frequency converter

The optics sets used in the frequency converter are type I KDP and type II KDP crystals arranged in a I/II phase matching configuration. The initial fundamental wave is linearly polarized along the ordinary axis of the doubler (type I), producing the fundamental and second harmonic waves that polarize along the extraordinary and ordinary axes of the tripler (type II), respectively. Finally, the third harmonic wave is generated by polarizing the wave along the extraordinary axis of the tripler, as shown in Fig. 1.

2.2. Mounting configuration

The optics set is a block with physical dimensions of 430 mm × 430 mm × 12 mm, and the frame is a windowed structure with overall dimensions of 430 mm × 430 mm × 10 mm. The optics set is supported on the frame at the contact area, which is 5 mm wide along the four edges of the bottom surface. The bottom surface of the optics directly contacts the top surface of the frame, and the right (+X) and downward (−Y) laterals of the optics are constrained to the X and Y directions, respectively. The bottom surface of the frame is rigid, and the normal of the mounting configuration is the Z axis. Gravity acts at a 45 degree angle to the normal of the YZ plane. The external load is applied to the loading area (the 5 mm wide area along the four edges of the top surface, facing the contact area on the frame), as shown in Fig. 2.

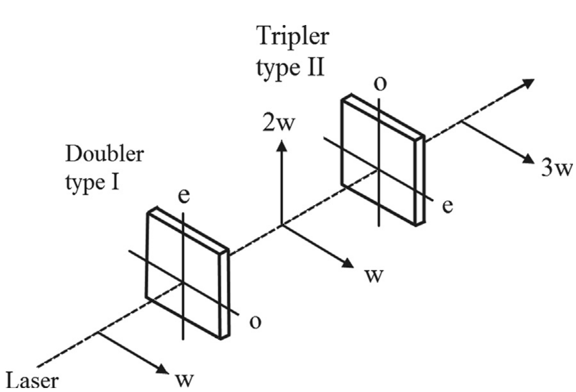

Fig. 1. Optical configuration for the frequency converter.

2.3. Finite element modeling

The mounting configuration is modeled and mechanically analyzed using the finite element package ANSYS. Because the machining accuracy of the optics is better than that of the frame, the surface of the optics is assumed to be smooth. The top surface of the frame is assumed to be a curved surface with a certain flatness, whereas the other surfaces are assumed to be smooth. To build the curved surface, key points with a Gaussian height distribution are established prior to using a spline curve to connect these points. An additional three straight lines are used to enclose the cross section, which is then extruded to form the solid frame structure. The curved surface is shaped as the top surface of the frame at the same time by repeating this process four times. The phase-matching and azimuth angles of the optics are set to 41.19 and 45 degrees, respectively. The optics and the frame are meshed with SOLID elements, and contact pairs are established between the bottom surface of the optics and the top surface of the frame with contact elements. The friction coefficient is set to 0.15. These parameters are used to create a model that contained 32317 nodes and 23548 elements. The right (+X) and downward (-Y) laterals of the optics are constrained to the X and Y directions, respectively, and the bottom surface of the frame is fully constrained. Gravity is applied at a 45 degree angle to the Z direction in the YZ plane, and the external load, in the form of pressure, is applied to the loading area, as shown in Fig. 3.

3. Mechanical and optical analysis theory

The mechanical and optical analyses are carried out consecutively. The distortion and stress are obtained from the mechanical analysis, and the phase mismatch and induced frequency conversion efficiency decrease are calculated from the optical analysis.

3.1. Mechanical analysis background

The mechanical analysis proceeds using the finite element method, in which the solid structure is divided into nodes and elements. The fundamental solution is calculated as the nodal deformation, and the stress is obtained based on this deformation. The deformation is solved according to the following equation:

$$[K]\ \{U\} = \{F\}, \tag{1}$$

where $\{U\} = \{\ u \quad v \quad w\ \}^T$ and u,v and w are the deformations in the X, Y and Z directions, respectively. The variable $[K]$ is the stiffness matrix and $\{F\}$ is the load matrix; these matrices are determined by the finite element model.

Based on the solved deformation, strain, an intermediate variable, is obtained as follows:

$$\{\varepsilon\} = [B]\ \{U\}, \tag{2}$$

where $\{\varepsilon\} = \left\{\ \varepsilon_{xx} \quad \varepsilon_{yy} \quad \varepsilon_{zz} \quad \varepsilon_{yz} \quad \varepsilon_{xz} \quad \varepsilon_{xy} \right\}^T$; ε_{xx}, ε_{yy}, ε_{zz}, ε_{yz}, ε_{xz} and ε_{xy} are the strains in X, Y, Z, YZ, XZ and XY directions, respectively, and $[B]$ is a geometric matrix defined as:

$$[B] = \begin{bmatrix} \frac{\partial}{\partial x} & 0 & 0 & 0 & \frac{\partial}{\partial z} & \frac{\partial}{\partial y} \\ 0 & \frac{\partial}{\partial y} & 0 & \frac{\partial}{\partial z} & 0 & \frac{\partial}{\partial x} \\ 0 & 0 & \frac{\partial}{\partial z} & \frac{\partial}{\partial y} & \frac{\partial}{\partial x} & 0 \end{bmatrix} \tag{3}$$

Stress is then solved for using the strain as follows:

$$\{\sigma\} = [D]\ \{\varepsilon\}, \tag{4}$$

where $\{\sigma\} = \left\{\ \sigma_{xx} \quad \sigma_{yy} \quad \sigma_{zz} \quad \sigma_{yz} \quad \sigma_{xz} \quad \sigma_{xy} \right\}^T$ and σ_{xx}, σ_{yy}, σ_{zz}, σ_{yz}, $\sigma_{xz}$and σ_{xy} are the stresses in the X, Y, Z, YZ, XZ and XY directions, respectively. The variable $[D]$ is the elastic matrix. For anisotropic materials, such as the KDP crystal studied in this paper,