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REDESIGN OF AN INDUSTRIAL LASER CUTTING MACHINE'S GANTRY IN COMPOSITE MATERIAL

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Abstract

This work is focused in the design stage of a composite structure intended to replace a metallic critical component in a 2D Industrial Laser Cutting Machine (ILCM). The component is the gantry, largely responsible for most of the ILCM's characteristics. These include precision and maximum acceleration, which are critical. The dimensioning of the component is initially performed based on analytical models, but latter stages use the numerical capabilities of Finite Elements Method. In the end it is possible to take advantage of the higher specific rigidity of composite materials to increase the maximum acceleration that the machine allows for while maintaining the precision.

1. Introduction

Industrial Laser Cutting Machine (ILCM) have has basic function the production of a workpiece with required geometry, with an acceptable surface finish and accuracy and at a rate as high as economically possible [1]. This rate, or productivity, is, in the case of high speed processing machines such as the ILCM, often primarily related to high acceleration capability. Therefore the ILCM must be capable of maintaining high precision while subjected to high accelerations [2]. The gantry is a highly critical machine element in this aspect, amongst others. Therefore, the design of a new gantry must be capable of generating a part with high stiffness. The material in which the new gantry will be produced must allow for adequate dynamic characteristics. This encompasses allowing for high stiffness whilst keeping low mass and good damping characteristics. In both fields (specific stiffness and damping) multilayered polymeric matrix composites (composites) have been proven to be more attractive than metals and other polymers. In fact, multi-layered polymeric matrix composites are commonly used for weight reduction and increasing the strength and rigidity of machine components [3]. This work shows the development of a new gantry that aims at ensuring the high precision the current machine can offer while allowing for a higher production rate.

1.1 Machine description

A schematic depiction of the Industrial Laser Cutting Machine (ILCM) is portrayed in Figure 1.

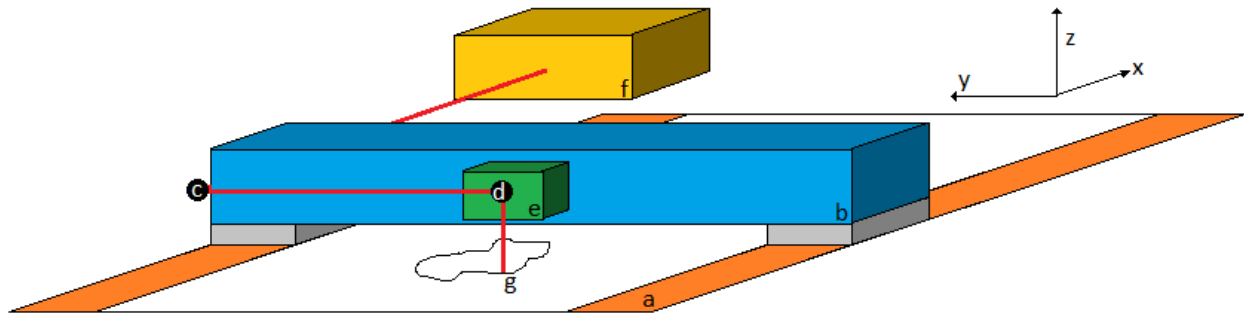


Figure 1. Schematic depiction of the ILCM and the cutting process. a) supporting structure; b) gantry; c) stationary optical components; d) movable optical components; e) cutting head; f) laser source; g) cutting spot.

As illustrated, the ILCM under analysis consists of three different main components: supporting structure (a), gantry (b) and cutting head (e). Attached to the gantry, on one of its ends, are the stationary optical components (c). These components project the laser beam to the movable optical components (d) that are contained in the cutting head (e). All these components have one main purpose: to ensure that the laser beam follows a path that takes it from the laser source (f) to the point where it must hit the target to allow for the production of the desired piece or cutting spot (g). The gantry moves along the length of the supporting structure and the movable components of the optical system move along the gantry. The combination of these orthogonal movements allows for the covering of the two dimensional space defined by the target (or part that is to be cut). The gantry is, therefore, a critical component in terms of the cutting performance of the ILCM. The gantry movement is produced by linear motors that allow for high placement precision and accelerations. The rigidity of the gantry must be such that it ensures deformations will not cause that the spot where the laser effectively hits the target distances more than the machines accuracy from the cutting spot.

1.2 Gantry characterization, Loads and Boundary Conditions

The current machine's gantry presents essentially a rectangular tubular section. The current component's visible part is the exterior shell, produced in metal sheet of uniform thickness. This shell has both aesthetic and functional purposes. Contained by the shell are several reinforcements. The gantry is designed in such a way that, despite the forces due to the accelerations required for its displacement, it presents minimum deformation. At the same time, it needs to be as light as possible to allow for low inertia. The gantry is subjected to several loading cases, depending on the acceleration that it suffers and the location of the components. The one that has been identified as critical is the loading case in which the gantry suffers maximum acceleration and the cutting head is in the center, while the cutting head is also subjected to maximum acceleration. In all load cases there are also other concentrated loads that come from the interaction of surrounding components. All of these loads need to take into account the acceleration the system is subjected to. They were taken into account in the following analysis but they will not be specified in this paper. The forces to be considered in this loading case are presented in Table 1.

Description of Loads	Effects of Loads
The weight of the gantry when subjected to the gravitational acceleration in the z direction.	Deflection in z axis
The load applied on the gantry by the cutting head when subjected to the gravitational acceleration in the z direction.	Deflection in z axis Torsion (rotation in the y axis) due to decentered load
The weight of the gantry when subjected to the acceleration in the x direction.	Deflection in x axis
The load applied on the gantry by the cutting head when subjected to the acceleration in the x direction.	Deflection in x axis
The load applied on the gantry by the cutting head when it is subjected to the acceleration in the y direction for the cutting head's movement.	Torsion in the z axis

Table 1. Description and effects of the Loads to be considered in the gantry's analysis.

2. Description of analysis

The dimensioning of the new composite structure had, as goal, the conception of an alternative component that, while capable of insuring the same degree of precision, would allow for higher accelerations. A 50% increase on maximum acceleration was defined as goal, as to allow for a new maximum acceleration of 3 g. The external dimensions of the composite structure are to be the same as the current component, as to ensure compatibility with the current system.

2.1 Analysis of the macro behavior

The analysis of the macro behavior of both the metallic and composite structures considered them as cantilevered beams. This way it was possible, by applying either a load or a moment at the edge, to assess the Moment of Inertia (I) or the Rotational Moment of Inertia (J) of the structure. This allowed for the determination of the composite layup that allows for equivalent bending and torsional behavior while minimizing the structure's weight. The first step was to study the current gantry.

2.1.1 Bending and torsional behavior of the cantilevered current metallic part

The analysis of the current metallic part was performed using Finite Elements Method (FEM). The software used was Abaqus/CAE 6.11-1. The geometry of the current metallic component was obtained and, in order to reduce computational cost, both external shell and interior reinforcements were considered as a single part. This approximation requires no contacts to be defined and assumes rigid connections between different parts. This is very similar to what is verified in reality. A midplane mesh was generated in ANSA pre-processing software. The powerful meshing capabilities presented by this software allowed for a better discretization of the domain, thus leading to more reliable outcomes from the FEM software. The part was then assembled as a cantilevered beam, having one edge encastrated and a load or moment applied to the other extremity. Loads were applied along the x and z direction and Torsional Moments around the y axis. The applied load was of 980 N and the torsional moment of 5.0E+7 N/mm.

The end result of this stage is presented in Table 2. Here are presented the maximum allowed deflections and rotations that the beam can display when subjected to the abovementioned loads.

2.1.2 Calculation of equivalent dimensions for a non-reinforced composite structure

This step consisted in the definition of the thickness of composite material required to produce a part that ensures equivalent machine performance. Because of the loads described in Table 1, two fiber orientations were selected for the design of the gantry. The unidirectional fibers are placed along the y direction and will be able to give higher bending rigidity. $\pm 45^\circ$ fibers are used for their higher Shear Modulus. The dimensioning performed at this stage considered the thickness of each of the two layers as a continuous variable. The calculations were performed considering that only the unidirectional fibers would be subjected to the bending loads and that the rotation of the beam would only be dictated by the thickness of the $\pm 45^\circ$ fibers layer. The calculations were also performed considering that the unidirectional layer will be the external one.

The loads applied to the composite were the same but, because the composite component is subjected to higher acceleration, the admissible deflections need to be recalculated. The maximum deformations admissible for the composite gantry are shown in Table 2.

Deformation	Maximum allowed value for metallic component	Maximum allowed value for composite component
Deflection in z	0.618 mm	0.618 mm
Deflection in x	0.321 mm	0.214 mm
Rotation in y	0.0199 rad	0.0133 rad

Table 2. Maximum allowed deformations for the composite gantry.

The calculation allows for the selection of the combination of four thicknesses that ensures the desired rigidity while, at the same time, minimizing the weight of the structure. Table 3 shows the different layers considered.

Layer Denomination	Description
$L_{uni,xy}$	Layer of unidirectional fibers in faces parallel to the xy plane
$L_{\pm 45,xy}$	Layer of $\pm 45^\circ$ fibers in faces parallel to the xy plane
$L_{uni,yz}$	Layer of unidirectional fibers in faces parallel to the yz plane
$L_{\pm 45,yz}$	Layer of $\pm 45^\circ$ fibers in faces parallel to the yz plane

Table 3. Different layers considered for the design of the composite component.

The first step was to determine the combination of $L_{uni,xy}$ and $L_{uni,yz}$ that allow for deflections in both x and z below the maximum ones established in Table 2 with a minimum weight. This was performed by iteratively considering shorter intervals between maximum and minimum values for each of the layers. Each interval was then subdivided to account for variations of 5%. A table was created having the variation of one of the layer's thickness in the vertical direction and the other thickness in the horizontal one. The solution that presented lower weight and admissible deformations would be used to define the shorter interval to further analyze its surroundings. Several initial intervals were studied as to try to avoid the convergence towards a local minimum. The process was stopped when the thickness level reached the 2 decimal places. The values were then used to calculate the external dimensions of the $\pm 45^\circ$ layer. The process was repeated to calculate the thickness of the fibers with higher Shear Modulus. Two alternative materials were tested: Glass Fiber with Epoxy Resin (60% Fiber Volume) and Carbon Fiber with Epoxy Resin

(also 60% Fiber Volume). The properties were gathered from [4]. The results for both materials are gathered in Table 4.

Layer Denomination	Carbon Fiber	Glass Fiber
$L_{uni,xy}$	7.37	24.18
$L_{\pm 45,xy}$	11.78	18.19
$L_{uni,yz}$	3.26	12.51
$L_{\pm 45,yz}$	12.16	18.36
Weight	76.82 kg	210.18 kg

Table 4. Minimum layer thicknesses that ensure enough rigidity for both Carbon Fiber and Glass Fiber. Weight for the design gantry is also displayed.

Due to the huge difference in terms of weight between the two solutions, the Glass fiber solution was dropped. This decision was also based in production difficulties that arise from the required thicknesses, as well as the fact that the gantry would be much heavier than the current metallic solution, (roughly 100 kg).

2.1.3 Comparison between analytical and numerical models

The numerical model was assembled in an analogous way to the simulation that allowed for the initial assessment of the current metallic component. The structure was also represented by shell elements and one of the edges encastrated. The load or torsional moment was applied to the other edge. The thickness of the two layers previously obtained was applied to the numerical model, as well as the same mechanical properties that allowed to their determination. The results obtained from both this step as well as the previous one are displayed in Table 5.

	Maximum Allowed deformations	Analytical model deformations	Numerical model deformations
Deflection in z	0.618 mm	0,6175	0,3369
Deflection in x	0.214 mm	0,2139	0,21325
Rotation in y	0.0133 rad	0,01326	0,0147

Table 5. Comparison between the deformations according to the analytical and numerical models.

As it can be seen in Table 5, the models are in agreement except for the deflection in z. The numerical model shows that the structure is suffering very little deformations under this load compared to what it could be. The next step will focus in ensuring that this overdimensioned structure can have its mass reduced while still ensuring sufficient rigidity.

2.1.4 Thickness adjustment of composite material

The first step to assess the possibility of weight reduction was to lower $L_{uni, xy}$. In order to do this, the section defined in the numerical model is varied. Figure 2 shows that, as this thickness lowers, the maximum deflections in z increase as expected. It also shows that it is possible to lower the thickness down to 3 mm without compromising the maximum allowed deflection in z. The opposite effect that the reduction of $L_{uni, xy}$ has on the rotation in the y axis can be explained by the increase of the external dimensions of the $\pm 45^\circ$, thus leading to higher torsional resistance. The increase of

the deflection in x goes above the maximum allowed one. This is addressed by increasing $L_{uni, yz}$ (Figure 2).

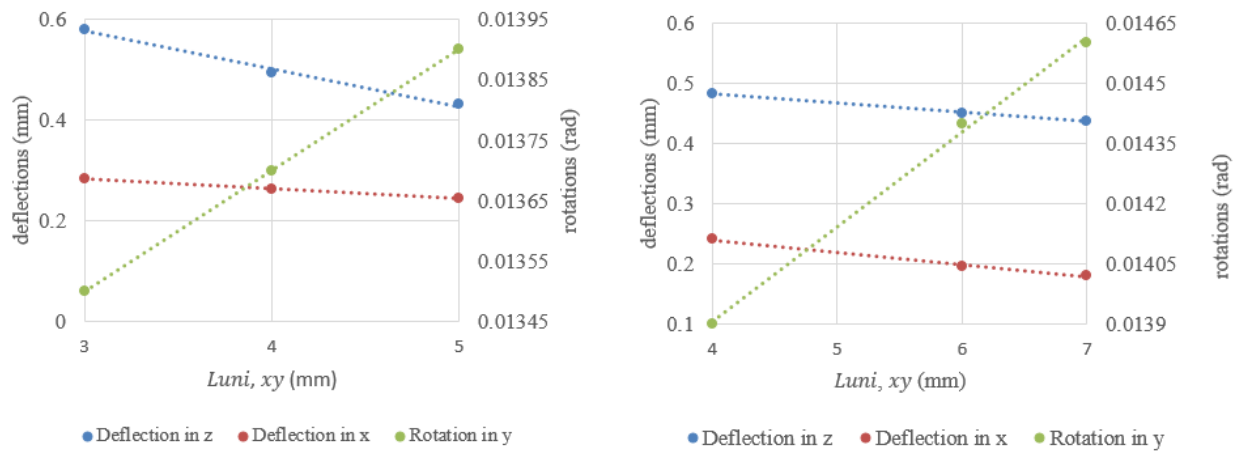


Figure 2. Effect of $L_{uni, xy}$ reduction in the deflections and rotations of the composite structure (on left) and .effect of $L_{uni, yz}$ increase in the deflections and rotations of the composite structure (on the right).

This analysis led to the definition of new values to $L_{uni, xy}$ (4 mm) and $L_{uni, yz}$ (6mm). The reduction of these thicknesses not only continues to ensure deflections bellowed the permissible level, but also improves the rotational behavior of the component and allowed a weight reduction of 17.2%.

2.2 Analysis of behavior on realistic load conditions

The previously generated geometry is now subjected to loading conditions that are representative of the machine's working conditions, as described in Table 1. Other components that apply more concentrated loads are also taken into account. These components are not geometrically represented in the numerical model. Instead a point, that has the same coordinates as their center of gravity, is represented. To these points are applied the loads that are equivalent to the mass of the component they are representing multiplied by the accelerations they are subjected to. Each point is subjected to loads that can have components in all three directions. The connection between center of gravity and the gantry are replaced by kinematic coupling. This way to model the connection between the gantry and surrounding components consists of a group of slave nodes (the nodes where the components are attached to the gantry) constrained by the translation and rotation of a master node (the point to which the loads are applied). Both the metallic and the reinforcement free composite structure were subjected to the same loads. For both of them, the considered outputs were the location of the point of maximum displacement and the displacement (both translational and rotational) of the point that represents the cutting head. This point was selected since it is both the endpoint of the optical system and the source of larger loads applied to the gantry.

The first step was to subject the metallic part to the loads that mimic the gantry's working conditions. Figure 3 shows the outcome of the numerical analysis of the gantry.

2.2.1. Behavior of composite structure and introduction of reinforcements

The initial results from the numerical model in which the composite model was subjected to loads portraying the working conditions showed concentrated deformations near the interface with some components. These working conditions take into account the higher accelerations that the composite gantry is subjected to. The approach selected to tackle this was the introduction of ribs along the axis of the gantry. These ribs increase the rigidity of the shell structure by transferring the loads that are applied to one face to the opposite ones. The material selected for these to be produced with was carbon fiber at $\pm 45^\circ$. The spacing between ribs was determined the established rib thickness and the punctual loads the gantry is subjected to.

3. Results

It was possible to compare the outcome of the two models. Figure 3 shows the outcome of the FEM simulation regarding the metallic component.

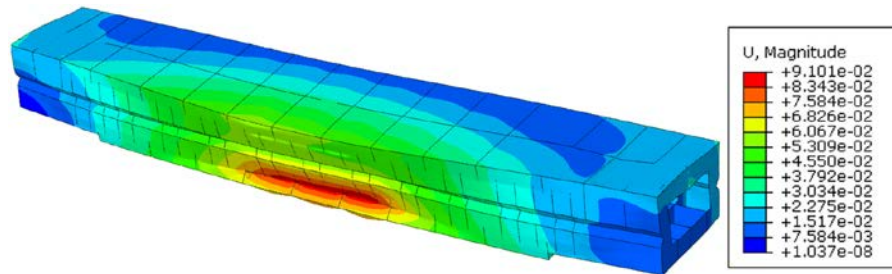


Figure 3. Outcome of FEM simulation for the metallic gantry subjected to realistic loading conditions. The color display the translational displacement magnitude. The deformation is magnified by 1400 times.

The area with higher displacement matches where the cutting head is attached to. In Table 6 is possible to see the displacement suffered by the cutting head's center of mass.

The result of the reinforced composite structure shows a deformation profile similar to that of the metal structure (Figure 4), as well as a cutting head displacement under the levels presented by the metallic structure (Table 6).

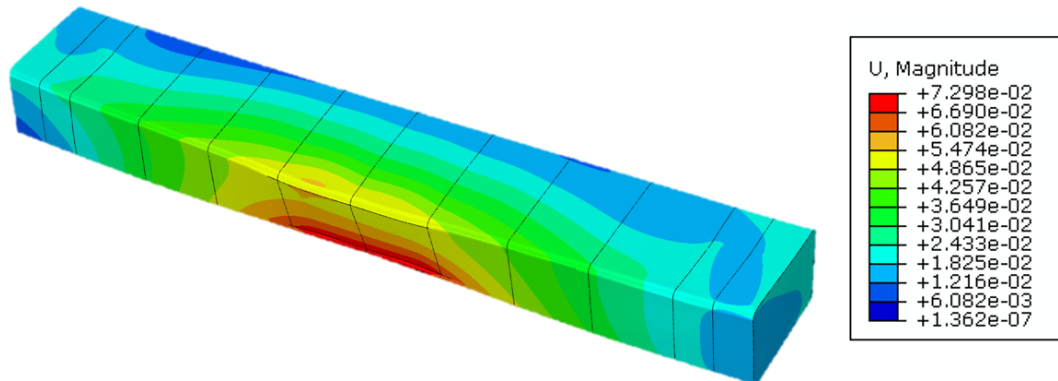


Figure 4. Outcome of FEM simulation for the composite gantry subjected to realistic loading conditions. The color displays the displacement magnitude. The deformation is magnified by 1400 times.

Displacement of Gravitational Center of Cutting Head	Metallic Gantry	Composite Gantry
Translation in x (mm)	0,0126	0,01422
Translation in y (mm)	0,0951	0,0799
Translation in z (mm)	-0,1190	-0,1189
Rotation in x (rad)	3,6160E-4	3,0881E-4
Rotation in y (rad)	-6,7109E-6	-4,4254E-6
Rotation in z (rad)	1,5886E-5	1,3141E-5

Table 6. Comparison between the displacement of the center of gravity of the cutting head on the metallic and on the composite component.

There is one value in the displacement of the center of gravity of the cutting head in the composite gantry that is higher than the one presented by the metallic one.

4. Conclusions

This paper describes the process followed to dimension a composite structure that presents similar rigidity as currently used metallic component. This process is based in analytical and numerical models and uses two steps in dimensioning. Overall agreement is achieved between these the two models. The macro evaluation is able to give an initial input that greatly shortens the dimensioning of a gantry subjected to realistic loading conditions. The performance of the beam subjected to these loads benefits greatly from the introduction of ribs perpendicular to the axis. Further efforts must be made to understand how relevant the slightly higher translation in x presented by the composite gantry is. The introduction of the ribs takes the weight of the composite gantry up to 81.3 kg. This means that the composite component is able to increase the maximum acceleration by 50% while keeping the same precision and decreasing the weight by roughly 18%. This value might change after the design for production stage.

Acknowledgments

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