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NEW METHODOLOGY FOR DESIGNING DIRECT-LASER-SINTERED MOTORCYCLE FRAME BASED ON COMBINATION OF TOPOLOGY OPTIMIZATION AND LATTICE IMPLEMENTATION

THANH BINH CAO & SLAWOMIR KEDZIORA

Research Unit in Engineering Sciences, University of Luxembourg, Luxembourg

ABSTRACT

The use of direct laser sintering (DLS) has become more attractive recently since it offers a promising tool in fabricating complex components rapidly. Particularly, the technique is seen more powerful when it is combined with computer-aid design and computational optimization. In spite of knowledge increment in the above areas presently, design method for sophisticated structures towards DLS is still far from being fully exploited. Therefore, this paper was issued to investigate a novel methodology of design, developed by combining topology optimization and lattice-beam implementation, for a blend-solid-lattice frame of a motorcycle. From the obtained results, it was recognized that the as-built tubular hybrid structure demonstrated comparable values of first resonant frequency and mass with respected to those of the original. Additionally, it was found that stiffness of the generated structures depended strongly on locations where lattice was substituted. In particular, less stressed frame's components were evidenced as appropriate regions for the substitution. The achieved results also revealed estimated buckling load factors, being circa 18 times higher than applied bumping loads acted on the tubular-lattice structure. Finally, equivalent stress predicted in static analyses confirmed all designs working safely in nominated conditions. Based on these achievements, it is believed that the new method worked quite acceptably in designing direct-laser-sintered motorcycle frame, and it is very promising to further develop the method as well as extend it into different complex direct-laser-sintered elements designed for future applications.

KEYWORDS: CAD/CAE, FEA, Direct Laser Sintering, Topology/Lattice Optimization

1. INTRODUCTION

Environmental pollution and Global Warming have become serious problems in the recent decades due to an increase of Carbon emission into the air. One of the main reasons was originated from an enhancement of the number of personal vehicles each year, including motorcycle. As a consequence, not only our environment and mankind health, but also the economy system was negatively influenced. The aforementioned circumstance placed great inquiries for researchers and engineers on how to further develop eco-friendly methods of designing transportation vehicles, so that, on one hand, it helps to increase reliability of the vehicles. On the other hand, it may contribute to a reduction of energy/fuel consumption during fabrication stage or during their working life.

To solve the above problems, development of a new design method should be concentrated initially. Currently, the uses of computer-aid design and computational optimization are becoming more attractive in design areas since they offer powerful tools to design complex components efficiently-and significantly speed up the process. Concerning these aspects, there have been several studies focused on optimization of motorcycle frame recently. For instance, Ballo et al., 2014hasattempted to develop a frame structure of a high performance motorcycle. By Modelica, they have constructed a multi-body model of the motorcycle for simulation of braking. Subsequently, Optistruct was employed to generate an optimized concept, of which total mass and structural compliance were minimized. With a similar design approach, Katawa (2015) concentrated his work on developing a frame of a racing motorcycle, showing high longitudinal stiffness(but low side stiffness)to be able to withstand high decelerating forces. In more recent days, AP Works (Airbus Group, 2016) have applied successfully a bio-inspired method on designing a frame of an E-motorcycle. By utilizing structural optimization during design step and Selective Laser Melting in manufacturing phase, the AP Works has built up an organic-exoskeleton-frame-like structure, demonstrating an adequate strength and a 30% reduction in weight with respect to those frames traditionally fabricated.

In terms of manufacturing tool, there has been increasing interest in applying DLS in mechanical industry presently. This is due to its promising ability of producing sophisticated components through a single fabrication step (Santos et al., 2006). Based on diminution of tooling, the DLS demonstrates advantages of lessening Carbon emission into the air and saving a considerable amount of fabricated materials. In view of these benefits, the DLS could be considered as one of promising techniques for eco-friendly manufacturing mechanical components, among which motorcycle frame is a suitable candidate.

Although motorcycle frame design has been developed for a certain long time, to the best of authors' knowledge, there is quite limited number of studies placed on designing motorcycle frame that can be produced by DLS. There is also very little research focused on performing implementation of lattice beams into the structure as well as on evaluation of how feasible the process is. Hence, this paper was issued to partly filling the "gaps" by investigating an innovative method of design applied on a motorcycle frame, based on topology optimization and meso-scaled lattice implementation. In the investigation, the "direct-laser-sintered" frame was modeled, analyzed, topology optimized, re-designed, and size optimized for a hybrid product development. A main objective of the proposed process was to generate a new blend solid-lattice structure, representing comparable stiffness and mass with respected to those of a realistic one, also taking into account aspects of aesthetic design and environmental protection.

2. METHODOLOGY

2.1 Original Frame Design and Analyses

CAD Geometry of Original Frame

CAD geometry of the original design was reconstructed based on a real frame structure of an Enduro sport motorcycle (KTM LC4 640 cc). To start with, the frame was firstly evaluated all realistic features to point out unimportant ones that were less influenced by stress. Then, these characters were eliminated during reconstructing for simplification. In the next step, manual measurements of main sizes were taken place for the frame. Subsequently, Autodesk Inventor was used to build up a concerned CAD frame, taking into account all possible feature eliminations as well as the measured sizes. It should be noted that within the reconstructed frame, welding was not essentially utilized to bond different components (e.g. upper brackets and upper tubes). Instead, a method of filleting common edges between these components was applied to create a homogenous solid structure. The aim of this modification was to create an appropriate geometrical reference for new frame designs, all of which were expected to be manufactured by DLS. The issued geometry was finally exported into a CAD file for constructing FE models and conducting FEM structural analyses.

FE Model of Original Frame

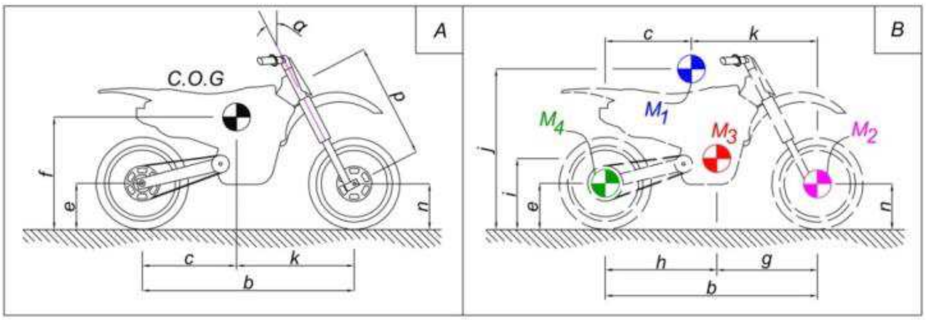
In order to build up an appropriate FE model for structural analyses, it was important to determine initially all factors that had high influences on the frame during working stages of the motorcycle. However, the higher number of factors was taken into account while developing the model, the more complicated construction process was. Therefore, to make the process simpler, different components of the motorcycle as well as a rider were substituted and merely represented by four concentrated masses(in the generated model).The replacements were demonstrated in Figure 1; whereas, significant dimensions of the issued sketch were defined and reported in table 1.

In the following step, load-cases acting on the motorcycle were determined. Practically, there has been a lot of working stages of a motorcycle which are essential to be considered when designing its frame. So far, to study all of these, huge efforts and time are required. Within the scope of this paper, only typical working scenarios of the motorcycle, including acceleration, braking, bumping, and cornering, were proposed and analyzed. These mentioned scenarios were simply represented in Figure 2.

Concerning acceleration (Figure 2-A), an accelerating factor (a) was determined by applying criteria of “Wheel-limited acceleration”, suggested by Cossalter (2002). According to the criteria, maximum acceleration might reach a value at which an absence of road reaction on the front wheel was detected. Equation (1) was used to calculate the acceleration. Other concerned parameters were referred to those stated in Figure 1 and table 1.

Table 1: Important Dimensions Represented in Figure 1

Denote	b	e	f	g	i	j	k	n	p	α
Corresponding value (mm or ^{0})	1510	300	750	750	400	1050	950	290	662	21



A: Motorcycle with its Centre of Gravity (C.O.G), B: Substituted Masses of Motorcycle

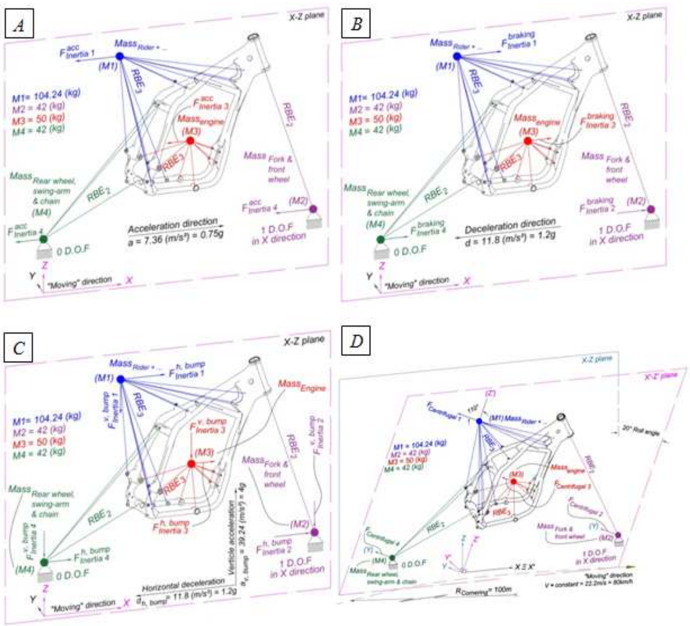
Figure 1: Motorcyle’s Layouts

$$a = \frac{c}{f} \cdot g \quad \textbf{(1)}$$

With a similar approach, deceleration (d), arising during braking (Figure 2-B), was identified by considering criteria of “Forward Flip Over” (Cossalter, 2002). In this state, the deceleration was upper limited by a value at which there was not a road reaction placed on the rear wheel. Equation (2) was used for this calculation.

$$d = \frac{k}{f} \cdot g \quad \textbf{(2)}$$

In terms of bumping (Figure 2-C), the scenario was supposed to occur instantaneously when the motorcycle hit, and tried to run through, a small obstacle on a road.Once bumping, reactions originated from the obstacle theoretically made the motorcycle to decelerate in horizontal direction, and accelerate vertically. To consider the bumping state critically, horizontal deceleration was supposed to be the maximal allowable deceleration shown in braking process, while vertical acceleration was assigned to a magnitude of 4g.



A: acceleration, B: braking, C: bumping, & D: cornering

Figure 2: Four Proposed Working Scenarios of the Motorcycle

Regarding the last load-case shown in Figure 2-D, the motorcycle was inspected when it was turning at a proposed constant velocity (V) of 80km/h. In this consideration, radius of curvature road (R) was assumed to be 100m; whereas, centrifugal acceleration ($a_{centrifugal}$) was determined following equation (3).

$$a_{centrifugal} = \frac{V^2}{R} \quad \textbf{(3)}$$

Based on the determined parameters, a FE frame model was then constructed by employing Altair Hyper mesh. During the construction, properties of steel (without specific label), used as a referenced material, were assigned to the 3D tetrahedral elements (TETRA10) of the model. Other essential setups of the model were carried out following a normal procedure of FE model construction for static analyses.

FEM Structural Analyses

To evaluate the issued FE model, modal and linear static analyses were essential to be carried out. Concerning the former, aims of the job were to estimate resonant frequencies of the frame at which it might vibrate resonantly, leading to detrimental damages, and to achieve as well predicted mode shapes. In such analysis type, an applied solver, Optistruct, was used to solve equilibrium equation (4)

$$[K - \lambda \cdot M]x = 0 \quad \textbf{(4)}$$

Where, K, M, and x were stiffness matrix, mass matrix and displacement vector, respectively, for eigenvalues (λ) calculation. Upon receiving the estimated eigenvalues, resonant frequencies (f) were computed based on equation (5) (Altair University, 2015).

$$f = \frac{\sqrt{\lambda}}{2\pi} \quad \textbf{(5)}$$

The three first frequencies of the analyses were subsequently extracted for further evaluations. In terms of the latter step, Optistruct was also employed to figure out static problems concerning the four determined load-cases. This task was handled by solving the well-known equation (6)

$$K \cdot x = f \tag{6}$$

Where, K was stiffness matrix, x was displacement vector, and f was external force vector. Finally, the predicted results of both analyses were saved into H3D files, which could be then visualized by Hyper-view to assess.

2.2 New Frames Design and Analyses

New Geometry Concept for Reference

To develop a new design concept, a new input geometry was firstly proposed and imported into Inspire software for subsequent setups. The input had to include constraint features as those of the original frame, which had to be kept during the developing phase so that a frame, created based on the output, could be used interchangeably. Moreover, the inputted geometry should have an enough design volume to provide high flexibility for the concept issuance. In principle, the Inspire will generate a novel concept of a design by solving a topology optimization problem (Altair, 2015). Dealing with this, a density method called Solid Isotropic Material with Penalization (SIMP) is applied to transform topology optimization into material density optimization. Actually, material density tends to vary in a range of [0, 1], demonstrating free volume state or fully dense material. During this optimization, however, the material density of elements has to be selected either as 0 or 1. Hence, a “Power law representation of elastic properties” algorithm, represented by equation (7), is subsequently employed to impose a penalty on intermediate density as well as assign mandatorily 0 or 1 as the density of each element.

$$\underline{K}(\rho) = \rho^{p} \cdot K \tag{7}$$

Where, $\underline{K}$ and K are penalized stiffness matrix and real stiffness matrix of element, ρ is the density, and p is the factor of penalization.

In the next step, pre-optimized model was established similar to that of the original frame, taking into account all necessary modifications, to synchronize the model with the proposed design volume. To control the optimization, boundary conditions, such as objectives, frequency and thickness constraints were essentially determined. In details, maximizing stiffness of output geometry was set as an objective of the optimization. The frequency was estimated towards maximizing its value; whereas, thickness constraints of the frame were restricted at 25mm. It should be also noticed that design space must be identified before the optimization. Upon completion of running, a novel concept of design was obtained, which would be used as an updated reference for redesigning frame geometry.

CAD Geometry of New Design Frames

Similar to process of reconstructing the original frame, redesigning solid and tubular CAD frames were also handled by employment of Autodesk Inventor as a developing tool. It was noticed that these redesigns should imitate as much as possible the features developed in the new design concept. Furthermore, additional elements were implemented into the redesigns as essential modifications aimed at reaching the pre-set targets. Those concerned CAD files, eventually, were exported for FE model constructions as well as FEM structural analyses.

FE Model of Redesigned Frame and FEM Structural Analyses

Issuances of new FE models, describing the redesigned frames working under different proposed conditions, were taken place similar to the process presented in 2.1. However, since there was presence of some modifications shown in the redesigned geometries, one should pay attention to differences in way of connecting four substituted masses and the redesigned frames. Concerning FEM structural analyses, a procedure used to analyze the new FE models was the same as that described in 2.1.

2.3 Frames with Lattice Structure, Design and Analyses

Implementation of Lattice Structure into Redesigned Frames

Basically, a process of lattice implementation was handled following the first stage of the lattice optimization. In such step, the lower density bound (LB) and the upper density bound (UP) must be firstly defined to help Optistruct identifying intermediate density elements. Subsequently, elements of which densities were lower than the LB will be eliminated while those with their densities higher than the UP would be assigned to solid elements. The remaining elements, therefore, possessed intermediate densities, which will be then transformed into a lattice structure. In addition, porosity of the FE models also has to be determined. This could be done via controlling the amount of intermediate density elements in the model (Altair, 2015). Since our main aim was to create structure, showing as high stiffness as possible, an output structure should be generated with low number of intermediate density elements, implying low porosity structure with minimal compliance. Stiffness of the obtained intermediate density elements would be approximately calculated following equation (8)

$$E = \rho^{1.8} \cdot E_o \tag{8}$$

Where, E was homogenized Young's modulus of intermediate-dense material, E_0 was Young's modulus of fully dense material, ρ was the density, and 1.8 was the natural penalty selected when low porosity was set.

It should be pointed out that to keep outer shapes of the redesigned frames for the aesthetic purpose the LB should be selected as zero. Besides, it is also necessary to notice that after running the lattice implementations, diameters of the issued lattice beams were directly proportional to density of the intermediate-dense elements (Altair, 2015). Since there were considerable amounts of tiny beams existed in the two FE models, which seemed unfeasible to be practically fabricated by DLS, an additional design step, which proposed by authors, was further added to substitute all lattice beams by those of which radii were set equally and fell in a feasible fabricated range. In this paper, these values were initially assumed to be 0.8mm for beams of both solid-lattice frame and tubular-lattice frame.

Size Optimization for Issued Lattice Structures and FEM Structural Analyses

Size optimizations were also handled by Optistruct to optimize cross-sections of the issued lattice beams. In our consideration, radii of beams' cross-section were set as design variables which could vary in pre-defined millimeter ranges of [0.3-1.3] and [0.2-1.3] for solid-lattice and tubular-lattice frames, respectively. Whereas, properties of lattice beams were considered as functions of the mentioned radii, principally described by equation (9), and could be modified to figure out problem of minimizing the lattice volume (Altair, 2015).

$$p = C_0 + \Sigma DV_i \cdot C_i \tag{9}$$

Where, p was the property, C_i were linear factors, and DV_i were design variables.

In addition to the above parameters, optimization's constraints were also necessary to be defined to guarantee that distributions of equivalent stress predicted in lattice structures of these frames would not exceed a value of 210MPa during different working scenarios of the motorcycle. It should be further noted that after the process, it was unnecessary to setup again the FE models. Once the size optimizations were completed, FEM structural analyses were then handled similar to the process shown in 2.1.

Linear-Buckling Analysis

In linear buckling analysis, the four proposed load-cases, used as references, were firstly analyzed in linear static mode, of which estimated stresses were applied to issue geometric stiffness matrix (K_G) (Altair, 2015). Buckling loads were subsequently predicted via Lanczos method by solving equation (10).

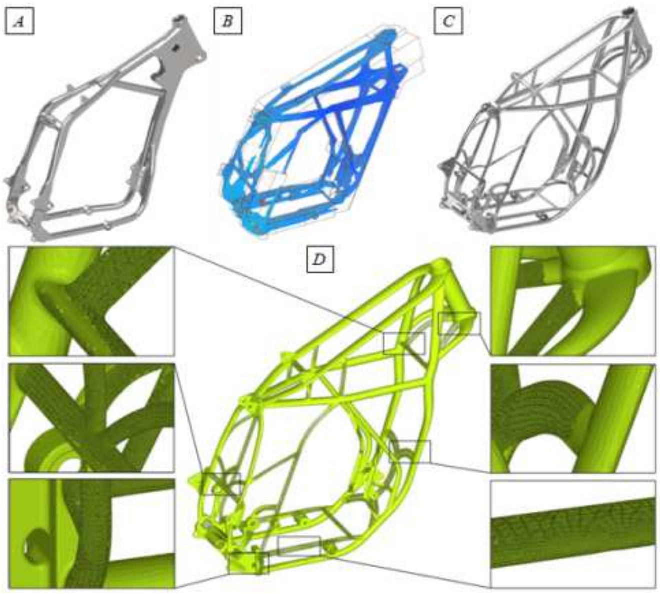
$$(K - \lambda \cdot K_G) \cdot x = 0 \tag{10}$$

Where, K was stiffness matrix of the frame containing lattice structure, λ was buckling load factor, and x was the related buckling displacement.

It is essential to emphasize that the lowest calculated eigenvalue was mainly dealt with the buckling. Additionally, the analyses provided information of buckling load factors (BLF), showing those estimated critical loads under which buckling would occur.

3. RESULTS AND DISCUSSIONS

Original Design, New Concept, and Redesigned CAD Geometries of Frame



A: Original Frame, B: New Concept, C: Redesigned Frame, D: Tubular Lattice Frame

Figure 3: Frame Design Development

In this section, original design, new design concept of the motorcycle frame, as well as those developed CAD geometries based on the optimized concept were represented. As can be seen from Figure 3, the original geometry was re-constructed towards a homogenous solid structure to adapt with the DLS technique. In Figure 3-B, the new concept of design obtained after topology optimization was represented. From the figure, it is seen that the proposed design volume was demonstrated by beam-type elements; whereas, a blue component indicated the achieved concept having constraint features that were not modified through all design stages. By referring to this concept, an initial redesigned geometry has been created and shown in Figure 3-C. It can be realized from Figure 3-C that the created geometry imitated almost all characters of the received conception, except for some disconnected elements that were necessary to be adjusted for generating well interconnected structure. In fact, this first developed frame was completely a dense object, being almost two times heavier than the original (table 2). Taking this point into consideration, final designs were further innovated by implementations of a hollow structure, lattice beams, or a combination of the two mentioned structures. Figure 3-D showed one of examples of the generated frames, in which lattice beams were implemented into the tubular structure.

Linear Static, Modal, and Linear Buckling Analyses

This section briefly demonstrated all concerned results of the linear static, modal, and linear buckling analyses. The obtained results were firstly summarized in table 2. From the table, it can be recognized that the highest equivalent stress values were estimated in bumping for all considered structures. This is probably due to the highest proposed static loads placed on the motorcycle system during this working condition. In particular, the most critical value of 204.6MPa that nearly reached yield strength of the referenced material (210.0MPa) was detected in the lattice beams of the redesigned solid-lattice structure. This value is likely the result of transmissions of the aforementioned loads to groups of tiny beams having small circular cross-sections (0.5 mm in radius). Whereas, maximal equivalent stress predicted in dense portion of the frame stood bit far below yield strength of the assigned material at 179.3MPa. Based on what we achieved, it is suggested that bumping was the most critical condition, which was necessary to be paid more attention to during designing process.

Table 2: Summary of Linear Static, Modal, and Linear Buckling Analyses' Results

			Original Frame	Redesigned Solid Frame	Redesigned Tubular Frame	Redesigned Solid-Lattice Frame		Redesigned Tubular Lattice Frame	
						Solid	Lattice	Solid	Lattice
Mass (kg)			13.1	25.2	16.5	5.69	15.19	12.72	2.42
Number of elements			1 058 908	1 191 879	1 738 218	432 614	998 632	1 334 541	549 414
Optimized R* (mm)						0.5		0.3	
Max static stress (MPa)	Acceleration		51.9	51.3	55.4	77.3	72.9	63.6	43.0
	Braking		118.9	69.1	104.9	108.5	172.5	107.6	97.4
	Bumping		167.9	120.8	140.7	179.3	204.6	170.1	83.5
	Cornering		83.8	35.3	63.0	68.2	74.8	77.6	38.2
Modal analysis	Mode 1	F*	129.6	133.4	146.3	71.0		119.9	
		T*	Torsion	Torsion	Torsion	Torsion		Torsion	
	Mode 2	F*	194.0	162.7	177.6	86.2		155.2	
		T*	Lateral	LT*	LC*	LT*		LC*	
	Mode 3	F*	219.7	228.9	231.2	118.0		171.3	
		T*	LT*	Lateral	Torsion	Torsion		Torsion	
BLF*	Acceleration					14.4		18.0	
	Braking					14.4		18.0	
	Bumping					14.4		18.0	
	Cornering					14.4		18.0	

*F denotes resonant frequency (Hz)
*T denotes type of resonant deformation
*LT denotes longitudinal tension
*R denotes radius of lattice beam after size optimization
*LC denotes longitudinal compression
*BLF denotes buckling load factor for lattice beams

It is also important to note that the mentioned yield strength of the referenced material was only proposed to support the frame-structural designs/comparisons, and it was seen as selective criteria when searching for suitable material for redesigned frame fabrications. Any metallic material expressing its yield strength higher than the criteria value may be seen as an appropriate candidate for the practical fabrication. In order to have clearer views of how stress distributed in original design as well as in the developed frames during bumping, estimated static stress results found in these structures were selectively collected and represented in Figure 4. According to the figure, it can be observed that high stress tended to distribute at locations where significant changes in cross-section of the frames were detected. For example, in the original design, the highest stress (around 168MPa) was found at joined areas between the two upper brackets and the two upper tubes, as well as at vicinity of the rear suspension hole. On one hand, occurrence of these high stress concentrations was derived from the above reason. On the other hand, high transmitted loads either from the heaviest mass M1 to the upper brackets (via RBE3, RBE-rigid body elements) or from the upper portion of the motorcycle to the “anchored” rear mass M2 (via RBE2) might result in the observed stress concentration. It should be explained additionally that the use of RBE2 in the FE models might lead to a lack of compliance producing artificial stress concentration in the components for which RBE2 were substituted. Nevertheless, an application of this 1-D element could provide an enough stable structure for linear static analyses as well as enable us to consider the frame under higher load interaction, meaning more safety consideration. Hence, to this extend the application of RBE2 could be accepted.

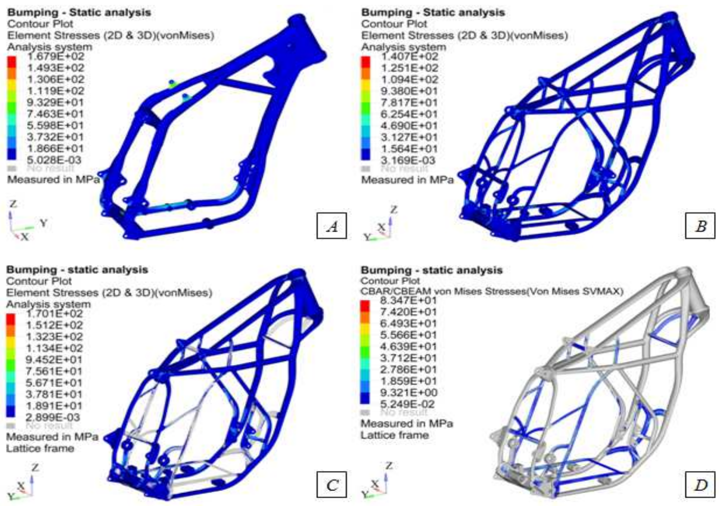


Figure 4: Equivalent Stress Distribution Predicted in: A: original design, B: redesigned tubular frame, C and D: redesigned tubular frame with lattice

In addition to the mentioned features of stress, it can also be realized from Figure 4 that there was a slight difference of stress distribution between left and right parts of the frame. The reason might be originated from asymmetric design of the original frame, particularly at pedal positions where only one connection area was arranged on the right side of the frame; while there were two connecting holes setup on the opposed direction. Consequently, the predicted stress would slightly distribute in an asymmetrical way within the frame. Dealing with the redesigned tubular-lattice frame shown in Figure 4-C, D, it is necessary to mention that since less stressed solid components of this structure were exclusively substituted by lattice beams an upper limit of stress found in these beams (83.5MPa) was equaled to circa one-third of that predicted in beam elements constituting the other lattice frame. Remaining portion of the frame, however, possessed a maximal estimated value of170.1MPa, demonstrating a small deviation with respected to the highest stress (179.3MPa) theoretically calculated in solid part of the comparing lattice frame structure.

With a purpose of giving evaluations on how stiff these frames were, FEM modal analyses were additionally handled to obtain important information regarding natural frequencies and sorts of resonant deformation. From those achieved results listed in table 2, it can be recognized that the redesigned tubular frame was probably the stiffest structure since it possessed the highest first resonant frequency of 146.3Hz, which was around 17Hz higher than that of the original one. However, the stiffest frame also represented its mass exceeding 3.4kg to that of the original. Therefore, in the proposed method of design, the lattice implementation was introduced into the frame structures to lessen as much as possible their masses while being able to maintain their stiffness at reasonable high values for safety reasons. In more details, results of the tubular-lattice frame indicated its first resonant frequency and mass being 119.9Hz and 15.14kg respectively, which were seen as comparable values with those of the original frame designed by conventional methods. These results may be used to prove that the proposed method of design for the direct-laser-sintered frame worked quite acceptably, and it is very promising to further develop and selectively extend this method to state-of-the-art direct-metal -laser-sintered structures for future applications. Nevertheless, it is necessary to emphasize that since a main aim of the lattice implementation was to reduce the mass and to attempt holding the frame stiffness at reasonable values, it was recommended to implement this porous component only into less stressed structures. Otherwise, despite of obtaining the mass reduction purpose, the designed structure would experience a significant drop in stiffness, as similar to that shown in the redesigned solid-lattice frame (table 2) of which a majority of the solid parts were substituted by lattice beams.

It is necessary to note that during mass estimation in Optistruct there might be an existence of overlapped material at junction volumes between the beams; hence, calculated masses for the lattice frames were over-estimated. In spite of this estimation, the results might imply that an enhancement of frame stiffness is closely associated with an increment of mass, as a result of “penalization”.

In view of buckling, the linear buckling analyses were only carried out for the two lattice frames since their tiny beams (slender components) might be really sensitive to buckling under influences of compressive stresses. The concerned results were listed in table 2, providing information of BLF which used to demonstrate safety factor in buckling mode. As can be observed from the table, an estimated BLF of 14.4 was found in the redesigned solid-lattice frame, indicating that the frame could be loaded minimally more than 14 times of the static loads until buckling happens. Mean while, in case of the tubular-lattice frame, its estimated value of BLF was 18. Nevertheless, it should be recognized that in practice due to presences of structural/material imperfections, the fabricated frame might be buckled at a load being lower than the predicted values.

4. CONCLUSIONS

This article represented a new method of designing direct-laser-sintered frame of a motorcycle. Principle of the method was based on combination between the topology and lattice optimizations. In details, an initial design volume was firstly proposed for running the topology optimization towards maximizing frame stiffness. Upon completion of redesigning frames based on the optimized concept, the lattice beams were implemented into these structures aimed at lessening their mass while maintaining their stiffness at the reasonable high values. The results showed that stiffness of the new lattice frames were strongly depended on components where the lattice structure was used to substitute. For example, if the porous structure was implemented into high stressed parts, the obtained frame would experience a signification diminution of stiffness with respected to that of the original solid form. This finding seems to be contrary to what stated in a study of Rosen, saying that solid sections of a structure replaced by cellular component could be stiffened. Despite of a certain stiffness reduction, the final design of the tubular-lattice frame demonstrated the comparable values of first resonant frequency and mass with respected to those of the original designed traditionally. In addition to these results, the buckling analyses revealed that the estimated buckling loads were approximately 18 times higher than the proposed static loads applied on the same frame structure. Finally, the equivalent stress estimations of the considered frames suggested that all of these frames worked in safety conditions.

5. FUTURE WORKS

In future, the works would focus on further developing the design method to optimize the direct-laser-sintered structures in terms of stiffness, mass, fatigue, and shape. Additionally, it would also be interesting to extend the method to different mechanical components that are promising to be applied the DLS technique. Last but not least, integrations between the design method and the relative fabrication process should be further investigated as well to be able to determine effective procedures of manufacturing high-quality-direct-laser-sintered components for future applications.

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