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Optimization procedure for an I-beam crane subjected to yield and buckling criteria

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ABSTRACT

The optimization of structure design has great benefits on the quality and cost. The current work implements a general optimization procedure that can be used for different structural applications. To demonstrate the functionality of the procedure, a real life application of gantry crane was selected. The gantry cranes are commonly used for lifting and moving heavy weights in different mechanical engineering jobs so that lightweight optimization of cranes with high load capacity constitutes a more and more demanding optimization research. Custom Welded I-Beam gantry crane is selected for numerical examples. The crane is composed of three rectangular plates with the same length and different thicknesses and widths welded together by full penetration welds over the span length to form an I-Beam profile. The live load and the beam span are the only two imposed parameters. The thicknesses and widths of plates are to be optimized so as to have the minimum cross section area while respecting yield, buckling, deflection and fatigue criteria. An all mathematical formula procedure based on Timoshenko beam theory and Crane Manufacturers Association of America (CMAA) in combination with the Genetic Algorithm (GA) is presented and a MathCAD code is implemented and applied to find the optimal I-Beam cross section dimensions. Nine numerical examples are presented for 8, 12 and 20 m span cranes subjected to 10, 20 and 40 ton rated loads. It is noticed that the optimized I-section configurations always show narrow and thick lower flange, wider and thinner upper flange and tall and very thin web. The lateral buckling and the upper flange local buckling limits are reached for 9 over 9 cases, the web buckling limit for 6/9 cases, the yield and fatigue limits for 3/9 cases and the deflection constraint is never critical. An optimization using ANSYS Workbench software with a 3D Solid Finite Element model is also done, gives good agreement and thus confirms that the proposed procedure is efficient.

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1. INTRODUCTION

The cranes are commonly used for different mechanical engineering applications. With regard to their service capacity, cranes in real life engineering are basically classified into five main classes [1]: standby, light, moderate, heavy and severe service cranes. The overhead gantry crane type is widely used to serve small or medium duty jobs, like repair shop, buildings service or in a machine shop. Lightweight cranes with high load capacity constitute an important requirement of the industry. To achieve this requirement, a customized I-beam crane is an interesting optimization research. Even if standard I-Beam profiles are available, they are just limited to some standard dimensions which are often far from the optimized one. The optimized beam could reduce up to 10% of weight with respect to initial design using standard profiles [4].
Some researchers have worked on optimization of the box profile girders [2, 3, and 4]; their procedure had in general the same methodology but with different optimization tools. They investigate the same concept of weight-strength ratio using theoretical optimization routines backed up by Finite Element (FE) simulation.
Zuberi, et al [3], examined the effect of rolling load on welded box cross section crane girder in terms of buckling and compression stresses in the flange. The girder volume is considered as an objective function while the stress and deflection criteria are used as constraints of the design. A commercial nonlinear optimization application called Generalized Reduced Gradient solver (GRG), integrated as a part of MS-Excel software, was performed to give partly optimized design variables. These design variables are then input to ANSYS code which handles more accurate stress and deflection calculations for verification purposes and, if necessary, doing further optimization.
Xiaogang et al [2] developed a modified Ant Colony Optimization (ACO) with mutation-based local search technique and applied it to solve nonlinear optimization problems having discrete variables. The new algorithm, abbreviated ACAM, were used to determine optimal crane design parameters and found to be faster by about 20% compared to the genetic algorithm (GA) and by 11% compared to particle swarm algorithm (PSO). Furthermore, it always give global optimized solution, while the original ACO algorithm may stuck at some local solution and fail to go further.
LUI, et al [4], conducted a three-dimensional parametric FE study of a double trolley box-girder crane using ANSYS APDL tool in conjunction with a Matlab platform to handle parameters. A set of two different algorithms, arc length algorithm (ALA) and nonlinear stabilization algorithm (NLA), were used to overcome optimization failures. An existing crane subjected to a given load capacity has been examined and optimized. The numerical results of their work shown significant save in material weight, 16% compared to original design.
This paper extends the similar techniques to optimize custom I-Beam crane designs. A custom I-Beam crane is made by welding three rectangular plates having the same length (L) and different thicknesses and widths by continuous full penetration welds, see FIG. 3. The live load and the beam span are imposed, whereas the thickness and the width of each plate should be determined so as to have the minimum weight while respecting the

yield, buckling, deflection and fatigue criteria. The mathematical calculations based on Cranes Manufacturer Association of America (CMAA) design procedure and the Hybrid Genetic algorithm (GA) are used to find the optimal dimensions of cross section that satisfy the design constraints. A MathCAD platform is written to handle these calculations. In addition, a 3D-solid FE model is created; stress analyzed and optimized using ANSYS Workbench software.

2. DESIGN OPTIMIZATION PROCEDURE

The crane design optimization under yield, buckling, deflection and fatigue criteria requires highly sophisticated techniques to achieve desired results. Such techniques must deal with iterative schemes which need a programming language or a mathematical application such as MathCad. In the last two decades, the general trends of solving such problems were emphasizing on carrying out a mathematical solution, a FE solution or a math-FE combined solution. This combined solution can be conducted in two different ways [3]. The first one independently carries out both techniques with the same initial values and takes the most optimal results between them. The second one uses the output results of the mathematical solution as input values of a FE solution. The later method is used in the present study. The proposed procedure is illustrated in Fig. 1. This procedure starts with the model formulation, i.e. define design variables, objective function ... Then, after entering the data of crane, which are in our case the span length, the rated load and the material, the optimization Hybrid Genetic Algorithm (GA) the details of which are shown in Fig. 2 is performed to give the so called Math-Optimal design variables.

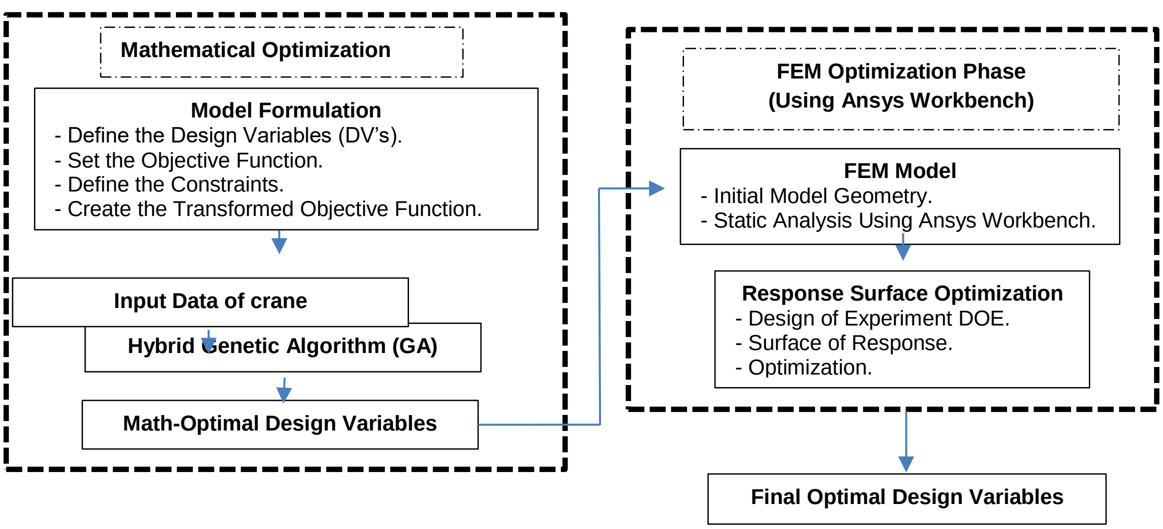

Fig. 1: Proposed Design Optimization Procedure

The Math-optimal design variables are input as initial variables to the FE Optimization phase using ANSYS Workbench 15 software in which the Response Surface Optimization method is used.

3. PROBLEM DESCRIPTION

The welded I-Beam crane and the loading conditions are shown in Fig. 3. The beam is formed by three plates that have the same length but different thicknesses and widths, joined together by continuous welds over the beam length. The dimensions and loading conditions are symbolically defined as follows:

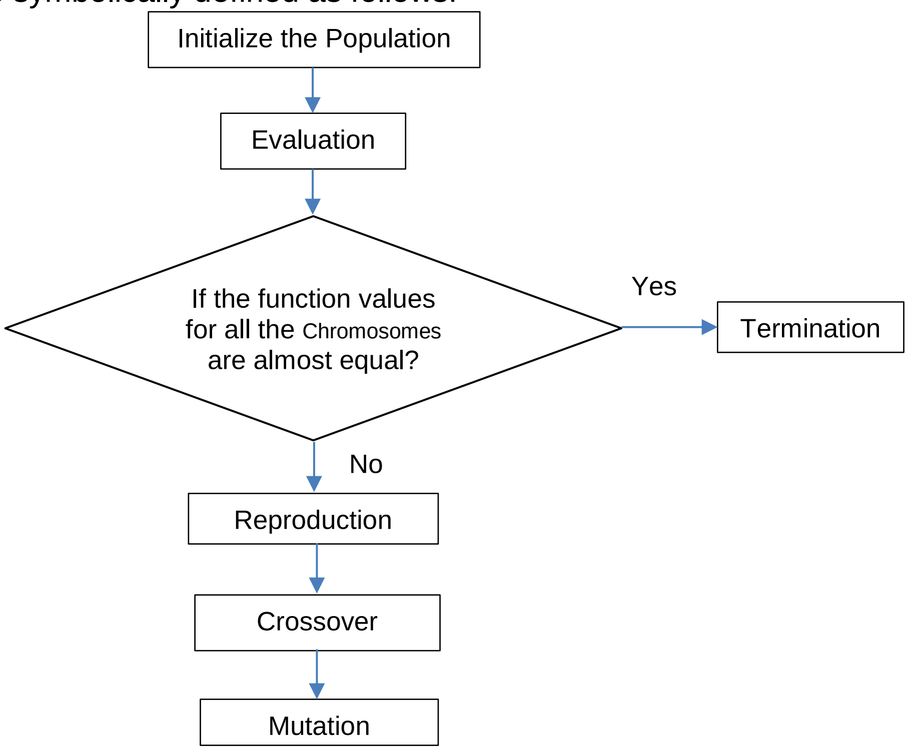

Fig. 2: Hybrid Genetic Algorithm [2].

b_1 : lower flange width, t_1 : lower flange thickness, b_2 : upper flange width,
t_2 : upper flange thickness, h : web height, t_3 : web thickness,
L : beam span, W_1 : crane weight, W_2 : live load (Lifting load),
x : distance of live load from left end

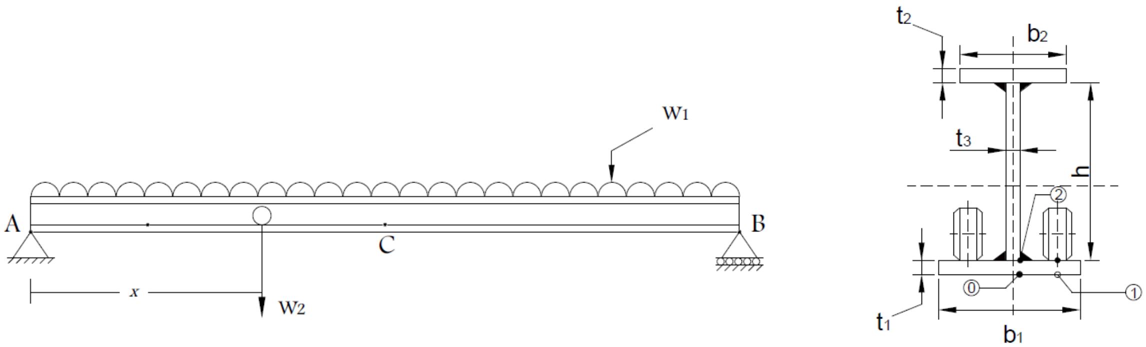

Fig.3: The Crane Beam Dimensions and Loading Conditions

The span length L, the lifted load W_2 and the material are the only three imposed data. The goal of the problem is to determine other geometric parameters so as to have a crane with a minimum weight while respecting all design constraints, which are described in two following sections.

4. OBJECTIVE FUNCTION

The beam span being fixed and the cross section constant, the weight is just proportional to the cross-section area so that the objective function is defined by the cross-section area, as follows.

$$f = b_1.t_1 + b_2.t_2 + h.t_3 \quad (1)$$

where the parameters b_1, t_1, … are shown in Fig. 3.

5. DESIGN CONSTRAINTS

The most important criteria of the Crane Manufacturers Association of America specification, known as CMAA74-2010, are considered and summarized as follows

- Tension stress Constraints (due to gravity and live load) :

  $\sigma_{\text{comb\_max}} - \sigma_{\text{Tallowed}} \leq 0$ (CAMM 74-3.4.4.1) (2)
- Lateral Buckling Constraint:

  $1.9 - f_{\text{Buckling}} \leq 0$ (Timoshenko Beam Theory) (3)
- Local Buckling Constraints:

  $h/t_3 - 260 \leq 0$ (AISC 2005 sec F13) (4)

  $b_2/2t_2 - 260/\sqrt{\sigma_y} \leq 0$ (CISC handbook p.p5-11) (5)
- Deflection Constraint :

  $\delta_v - L/600 \leq 0$ (CAMM 74-3.5.5) (6)
- Fatigue Constraint (due to repeated load fluctuation ΔW_2 only) :

  $(\Delta\sigma)_{\text{comb}_{\max}} - \Delta\sigma_{\text{allowed}} \leq 0$ (AISC 2005 sec F13) (7)

where $\sigma_{comb} = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x\sigma_y + 3\tau_{xy}^2}$ is the Von-Mises equivalent stress;
$\sigma_{Tallowed}$ = Allowable tension stress, according to CAMM 74;
δ_v = Maximum vertical deflection
$\Delta\sigma$ stands for stress range
σ_Y = Yield strength

$f_{Buckling}$ = Buckling load factor, which means a factor to be multiplied to all applied loads to produce linear buckling of the structure. This factor is initially given by the linear buckling theory, e.g. Timoshenko formulas or by a FE model. It is valid only if the linear buckling stress, which is $\sigma_{Cr0} = f_{Buckling*}|\sigma_{upper\ flange}|$, is less than ½*$\sigma_Y$; Otherwise, it must be modified to take into account the plastic deformation during buckling. The corrected critical stress is calculated using Johnson's empirical formula $\sigma_{cr} = \sigma_Y\left[1 - \frac{\sigma_Y}{4\cdot\sigma_{cr0}}\right]$, [6] and the corrected buckling load factor is given by $f_{Buckling} = \frac{\sigma_{cr}}{|\sigma_{upper\ flange}|}$.

6. OBJECTIVE FUNCTION TRANSFORMATION

The exterior point penalty function is used to transform the constrained optimization problem into an unconstrained problem. The general form of the transformed objective function is:

$$F(X, r_h, r_g) = f(X) + r_h\left[\sum_{k=1}^{i} h_k(X)^2\right] + r_g\left[\sum_{j=1}^{m}(max\{0, g_j(X)\})^2\right] \qquad (8)$$

where X is the vector representing the design variables, h_k is the kth equality constraint if any, g_j is the jth inequality constraint, r_h and r_g are two additional variables called penalty multipliers.

7. NUMERICAL EXAMPLES

Table 1. Crane Specifications according to CMAA 74-2010

Variable	Value/Units
Rated Capacity	10, 20 or 40 tons
Service Class D	Heavy Service
Load Class L_3	Normal load = 2/3 of rated load
Cycles Class N_2	Up to 500000 cycles
Span	8, 12 or 20 m
Trolley Weight	1 tons
Other equipment Load	1 tons
Bridge Wheel per rail	One on each side

Nine cases defined by three span lengths (8, 12 and 20 m) and three rated loads (10, 20 and 40 tons) are selected as numerical examples. The crane specifications are listed in Table 1. The material used for the crane is 350W structure steel with yielding strength $S_y = 350\ \mathrm{MPa}$, density $\rho = 7850\ \mathrm{kg/m^3}$, Young's modulus $E = 200\ \mathrm{GPa}$, shear modulus $G = 77\ \mathrm{GPa}$ and Poisson's ratio $\nu = 0.3$.
The mathematical optimization procedure described in section 2 is programmed using MathCad Code. The values of GA parameters are summarized in Table 2.

Table 2. Genetic Algorithm Parameters

Parameter	Used Value
Number of Variables	$N_V = 6$
Population size	$N_P = 120$
Probability of crossover	$P_C = 0.85$
Probability of mutation	$P_M = 0.05$
Mutation Parameter	$B_M = 5$
Maximum generation number	$G_{MAX} = 300$

8. FINITE ELEMENT MODEL

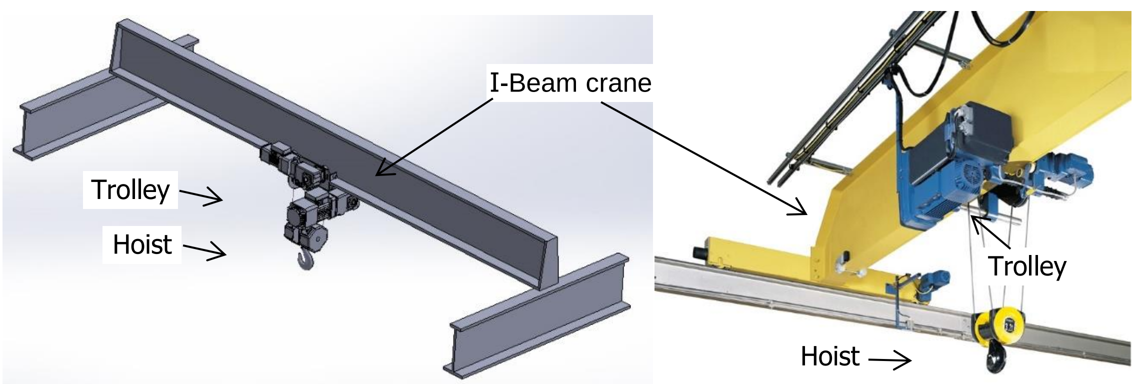

Fig. 4. Three dimensional images of the crane

The Fig. 4 shows a 3D drawing of an I-Beam crane, the Fig. 5 (a) shows the overall view of a 3D-solid FE model of the crane created in ANSYS Workbench © 15, and the Fig. 5 (b) shows a local zoom around the contact region between the lower flange and the wheels. This model contains about 28300 nodes, The lower edges at ends are vertically supported and the loads to be considered are composed of the distributed gravity load W_1 (weight of the beam) and the concentrated load W_2 applied on the wheels, W_2 being the combination of the lifted load, the weights of trolley and hoist. All loads are adjusted by factors according to CMAA 74 Specifications. The rated load plus gravity are applied when considering the yield and buckling constraints, inequalities (2) to (5), while the normal load fluctuation, which is just 2/3 of rated load without gravity, is applied when considering the deflection and fatigue constraints, (6) and (7). For the FE model, the Surface Response Optimization method, already integrated in Ansys Workbench, is used.

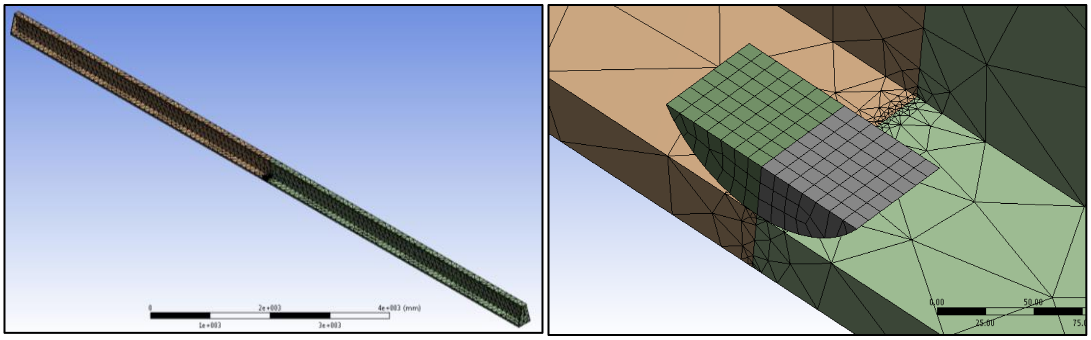

Fig. 5. Finite Element Model of the crane

9. NUMERICAL RESULTS

For reducing calculation time, it needs to input the reasonable lower and upper bound values of each design variable. The bounds used for all 9 cases are shown in Table 3.

Table 3. Lower bound and upper bound of design variables

DV's / Bounds	t_1	b_1	t_2	b_2	t_3	h
X_{low}	2	150	2	150	3	250
X_{upp}	100	600	100	600	100	1675

The optimized design variables for 9 cases are presented in the Tables 4.1, 4.2 and 4.3. The Table 4.1 results are for short span cranes with three different rated loads, the Table 4.2's are for intermediate span cranes and the Table 4.3's are for long span cranes.
It is noticed that the lateral buckling and the upper flange local buckling limits are reached for 9 over 9 cases, the web buckling limit for 6/9 cases, the yield and fatigue limits for 3/9 cases and the deflection constraint is never critical.

Table 4.1 Optimal Design variables and constraint parameters for 8 m cranes

L = 8 m	**10 tons**		**20 tons**	**40 tons**	**Bounds**
	MATH	FEM			
t1 (mm)	27.82	27.98	37.88	52.62	[2, 100]
b1 (mm)	150.01	150.0	150.16	150.04	[150, 600]
t2 (mm)	6.99	7.43	8.38	9.08	[2, 100]
b2 (mm)	194.20	186.1	220.18	252.14	[150, 600]
t3 (mm)	3.00	3.00	3.19	4.38	[3, 100]
h (mm)	608.64	650.88	826.46	1137.03	[250, 1675]
Area (m^2)	0.00736	0.00753	0.0102	0.0152	
σ_{com_max} (MPa)	221.9	201.9	224.9	225	<= 225 MPa
$F_{Buckling}$	1.9	1.96	1.9	1.91	>= 1.9
h/t_3	202.876	216.96	259.37	259.88	<= 260
$b_2/2t_2$	13.895	12.524	13.14	13.887	<= 13.898
δ_v (m)	0.0074	0.0063	0.0057	0.0043	<= 0.013 m
$(\Delta\sigma)_{com\_max}$ (MPa)	166	160.1	165.9	165.2	<= 166 MPa

Table 4.2 Optimal Design variables and constraint parameters for 12 m cranes

L =12 m	10 tons	20 tons	40 tons	Bounds
t1 (mm)	27.88	53.52	54.61	[2, 100]
b1 (mm)	179.54	150.01	206.13	[150, 600]
t2 (mm)	10.34	12.04	13.41	[2, 100]
b2 (mm)	287.15	334.55	372.14	[150, 600]
t3 (mm)	3.02	3.12	4.21	[3, 100]
h (mm)	785.98	811.78	1094.81	[250, 1675]
Area (m^2)	0.0104	0.0146	0.0209	
σ_{com_max} (MPa)	203.9	188.9	225	<= 225 MPa
$F_{Buckling}$	1.9	1.9	1.9	>= 1.9
h/t_3	259.93	259.98	259.98	<= 260
$b_2/2t_2$	13.884	13.896	13.877	<= 13.898
δ_v (m)	0.0095	0.012	0.0097	<= 0.02 m
$(\Delta\sigma)_{com\_max}$ (MPa)	163	121.3	164.1	<= 166 MPa

Table 4.3 Optimal Design variables and constraint parameters for 20 m cranes

<table>
<tr><th>L = 20 m</th><th>10 tons</th><th>20 tons</th><th colspan="2">40 tons</th><th>Limits</th></tr>
<tr><td>t1 (mm)</td><td>30.14</td><td>92.15</td><td colspan="2">54.41</td><td>[2, 100]</td></tr>
<tr><td>b1 (mm)</td><td>275.88</td><td>150.29</td><td colspan="2">347.13</td><td>[150, 600]</td></tr>
<tr><td>t2 (mm)</td><td>15.06</td><td>14.98</td><td colspan="2">18.1</td><td>[2, 100]</td></tr>
<tr><td>b2 (mm)</td><td>418.41</td><td>416.3</td><td colspan="2">501.7</td><td>[150, 600]</td></tr>
<tr><td>t3 (mm)</td><td>3.29</td><td>3.36</td><td colspan="2">4.72</td><td>[3, 100]</td></tr>
<tr><td>h (mm)</td><td>821.75</td><td>872.94</td><td colspan="2">1226.51</td><td>[250, 1675]</td></tr>
<tr><td>Area (m^2)</td><td>0.0173</td><td>.0230</td><td colspan="2">0.034</td><td></td></tr>
<tr><td>σ_{com_max} (MPa)</td><td>202.5</td><td>199.5</td><td colspan="2">218.2</td><td><= 225 MPa</td></tr>
<tr><td>$F_{Buckling}$</td><td>1.9</td><td>1.9</td><td colspan="2">1.9</td><td>>= 1.9</td></tr>
<tr><td>h/t_3</td><td>249.96</td><td>259.47</td><td colspan="2">259.95</td><td><= 260</td></tr>
<tr><td>$b_2/2t_2$</td><td>13.89</td><td>13.891</td><td colspan="2">13.896</td><td><= 13.898</td></tr>
<tr><td>δ_v (m)</td><td>0.025</td><td>0.032</td><td colspan="2">0.023</td><td><= 0.033 m</td></tr>
<tr><td>$(\Delta\sigma)_{com\_max}$ (MPa)</td><td colspan="2">157.7</td><td>94.61</td><td>166</td><td><= 166 MPa</td></tr>
</table>

It is also noticed that the optimized I-section configurations always show narrow and thick lower flange, wider and thinner upper flange and tall and very thin web. The Fig. 6 approximately illustrates the optimum I-Beam cross sectional configuration for a 20 m crane subjected to 20 tons lifted load.

If we compare the custom I-beam configuration of Fig. 6, for which the cross section area is $A = 0.023\ m^2$, to a doubly symmetrical I-beam, for which the optimized dimensions should be $t_1 = t_2 = 39.53$ mm, $b_1 = b_2 = 307$ mm, $t_3 = 3.85$ mm, $h = 996$ mm and the cross section should be $A = 0.028\ m^2$, it is found that the customized I-beam could save almost 18% of the weight.

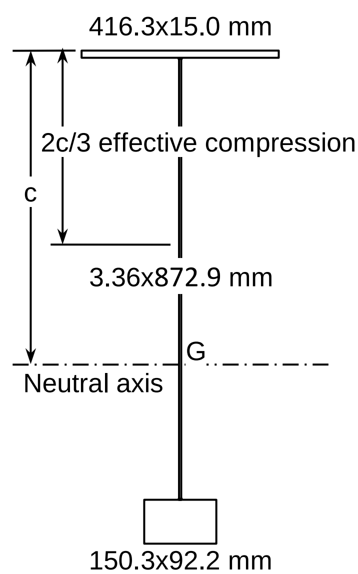

Fig. 6. Optimum configuration of an I-section

The design parameters given by the Math optimization are then input to a FE procedure using Ansys Workbench 15 with a 3D non-linear solid model due to the contact between the wheels and the lower flange. The Surface Response Optimization method in Ansys Workbench is used and subjected to the same constraints as before, excepted the linear buckling constraint, because linear buckling does not work with non-linear contact models. However, the buckling constraint ($f_{Buckling} \geq 1.9$) is replaced by an approximate constraint on the slenderness ratio against lateral buckling so as to give a comparable load buckling factor. This slenderness ratio is given by $\lambda = L/r_{cy}$ where r_{cy} is the lateral radius of gyration of the effective compression area which is empirically the 2/3 outermost of the compression side of the cross section (see Fig. 6).

FE stress calculation with non-linear contact and optimization procedure is very time consuming, we chose only one case to show FE results, which is the 8 m and 10 ton case. The slenderness ratio constraint for this case is $\lambda \leq 190$. The new optimized design parameters given by FE procedure are shown the FEM column of Table 4.1, which are slightly different but quite close to the Math results. The Fig. 7 shows that the maximum Von-Mises stress is in the lower flange right under the wheels.

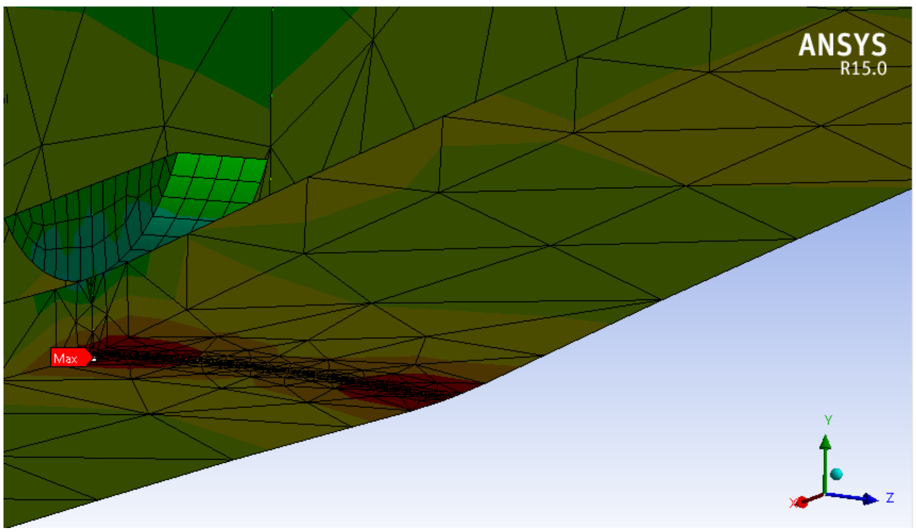

Fig. 7. Location of maximum Von-Mises stress

10. CONCLUSION

A Hybrid Genetic Optimization Algorithm (GA) and a Mathematical optimization procedure are programmed in MathCad and successfully applied to custom welded I-Beam cranes with different spans and rated loads subjected to yield, buckling, deflection and fatigue criteria. It is found that the constraints of general lateral buckling and local buckling of the upper flange are always reached for all cases, the web local buckling constraint is critical for about 66% of cases, the yield and fatigue constraints are critical for 33% of cases and the deflection constraint is not a problem at all. The optimized custom I-section has a configuration of narrow and thick lower flange, thinner and wider upper flange and tall-very-thin web which could save about 18% of weight compared to commercial standard I-Beam. FEM optimization using Surface Response method gives comparable results and confirms that the proposed procedure is efficient.

For future works, the FE optimization taking into account non-linear buckling due to contact or plasticity constitutes an important challenge. Furthermore, the optimization procedure with multi objective functions such as weight and cost will be also an interesting future work.

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