Design and Optimisation of Crane Jibs for Forklift Trucks

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Abstract— **The objective of this paper is to present a new methodology of calculation by means of the F.E.M. applied to crane jibs. This analysis has been carried out in terms of strength and stiffness, and for any type of crane jib: telescopic crane, lattice crane, closed beam crane, etc.**

There have been simulated different load cases and boundary conditions that the structure should bear. This load cases can be obtained from normal and extreme operation of the crane, and there have been taken into account current regulations and previous experience.

It has been also necessary to define the methodology to simulate non structural elements, contacts, materials, etc.

Concerning regulations, it has been necessary to make an adaptation of the regulation UNE-58536, because the specific regulation affecting these crane jibs (UNE 1726), was not enough restrictive. Moreover it has been necessary to estimate some translational and rotational velocities to apply the regulation.

Furthermore, the welding of some parts has been simulated and analyzed, and it has been carried out a comparative analysis between the experimental results and the numerical results for some load cases. Once this correlation has been obtained the methodology of calculation and the numerical results are validated.

***Index Terms*— jib, F.E.M., strength, crane, forklift truck**

I. Introduction

The article presents a methodology for the design and optimization of crane jib for manipulative telescopic using the finite elements method. For it, the design attends both to stiffness, strain, weight and regulation criterions and it is possible to apply to all types of crane jibs: telescopic (where the dynamic effects are relevant), in lattice, closed box, etc.

The start point of the analysis is the UNE-58536 standard because the 1726 one is insufficient and slightly restrictive for the calculation; it has been necessary to estimate some speeds.

From these standards and the previous experience of the manufacturer there have been established the cases of load and contour conditions, both for habitual use and for extreme use and they have defined the materials, the contacts between pieces, the welds, etc.

To check this regulation it has been applied to two types of crane jibs, a telescopic and a closed box one, which later have been optimized using numerical technologies and finally the telescopic jib has been tested using extensometrical procedures in order to check the numerical results and the methodology of design.

II. Associate Normative

For this type of crane jibs, they have not their own regulation, but, it is necessary to use the regulation relative to the fork lift one and, specifically, the standard UNE-1726: "*Carretillas autopropulsadas de capacidad hasta 10.000 Kg. y tractores industriales con un esfuerzo de tracción al gancho de hasta 20.000 N inclusive*", in that, all the structural and safety measurements that must support a self-propelled fork lift and its accessories are detailed.

With regard to the paragraph that is of interest in this norm (ap. 6.2), which indicates is: "The structural components of the fork lift and its accessories must support a static load of $1.33Q_1$ and $1.33Q_2$ during fifteen minutes.

- Q_1 it is the rated load at the normalized elevation and at the normalized distance of the centre of gravity, according to the indications of the capacities plate.

- Q_2 it is the real capacity at the maximum elevation height of elevation, according to the indications of the capacities plate.

As consequence of the test, it must not be visible any damage or permanent deformation"

On having observed this norm, it is possible to appreciate that it is a slightly restrictive and it is necessary to adopt for a correct design of our structure the regulation UNE-58536: *"Reglas para el cálculo de las estructuras de las grúas móviles de uso general"* applied to movil cranes mounted on tires.

III. Application of the Standard UNE-58356 in a Real Case

The standard UNE-58356 presents some load cases (Table 1 of the norm) and that are calculated as a combination of loads with a amplification coefficient (Table 5 of the norm):

Main loads:

Own weight of the jib (G)

Service loads (F): weight of the useful elevated load plus the weight of the accessories (F_0: block of pulleys, hook, cable, …). *Dynamical effects produced during the elevation and the descent of the load in service (Φ)*: This factor is calculated using the formula:

$$\Phi = 1.1 + 0.13 \cdot V_h(m/s) \quad (1)$$ with V_h <1.5 m/s

Φ= 1.3 for the other cases.

V_h is the maximum elevation velocity of each element (cable, load, …)

Forces due to the effects of the inertia of the movements of the crane: translation (T), rotation(S) and range (L): these forces are applied separately, both the own weight and the service load and it will be explained later.

Additional Loads

Wind loads in the most unfavourable position, both in service (W_i) –standard UNE 53-113- and out (W_o) delimited by the designer. They will be explained later.

Special loads: Load of static test of the norm UNE

58-501 in the paragraph 11.6. It is a overturn norm that indicates a load of 1.25 times the rated load.

With all these cases using the method of "partial coefficients of safety and tensions limits", the load cases that appear are the following ones, using the amplification coefficients of the table 5 of norm.

Normal load conditions

Case 1: normal load without wind

$$1.2 \cdot G + 1.35 \cdot \phi \cdot F + 1.5 \cdot T / S / L \qquad (2)$$

So it is necessary to calculate separately during rotation, translation and in increase of the range.

Case 2: normal load with wind

$$1.09 \cdot G + 1.2 \cdot \phi \cdot F + 1.35 \cdot T / S / L + 1.2 \cdot W_i \qquad (3)$$

Special load conditions:

Case 3: Out of service with wind.

$$1.09 \cdot G + 1.2 \cdot F_0 + 1.35 \cdot T / S / L + 1.9 \cdot W_0 \qquad (4)$$

Case 4: during the assembly with wind: this particular case is not going to pass never in this structure

Case 5: Static load case

$$1.09 \cdot G + 1.09 \cdot F_0 + 1.09 \cdot 1.25 \cdot (F - F_0) \qquad (5)$$

On having observed these load cases, it demonstrate that they are more unfavourable than the load cases established by the standard UNE-1726 and that contain to them.

On the other hand, for the calculation, it is necessary to establish the stress criteria, which, according to the norm, establish that:

Combined limit strain (using Von Mises' formula)

$$\sigma_{V.M,\max} \le \frac{\textit{Yield} \lim it}{1.11} \qquad (6)$$

Buckling calculus

$$\sigma_{compresion,\max} < 0.9 \cdot \frac{\textit{Buckling} \lim it}{1.11} \qquad (7)$$

Shearing strain calculus

$$\tau_{\max} < \frac{\textit{Buckling} \lim it}{1.92} \qquad (8)$$

Then, it is possible to proceed to the calculation of these elements using the diverse methodologies of structural calculation and materials resistance of materials.

IV. FORCES DUE TO THE INERTIA EFFECTS OF THE CRANE MOVEMENTS

The calculation of the forces due to the inertia, is more difficult to calculate and, they depend on every type of vehicle, of the brakes that it mounts, the hydraulic system, etc.[1]

Consequently, for a correct quantification of them, it would be necessary a test with accelerometers, but since it is not always possible, there will be tried to adapt the formula used for the dynamical factors, from the maximum speeds recommended in the regulation of cranes UNE-58-507-77 that establishes the table 1:

Table 1: Maximum admissible velocities for a crane

Maximun charge	Less than 10 Tn	Between 10 and 15 Tn	Between 15 and 20 Tn	Between 20 and 30 Tn	Between 30 and 40 Tn	More than 40 Tn
Translation velocity (m/s)	2,78	2,36	1,94	1,39	1,25	0,97
Rotation velocity (m/s)	0,37	0,31	0,25	0,25	0,21	0,21
Maximun velocity in the end with a range of 16 m	5,91	5,02	4,02	4,02	3,35	3,36

Maximun charge	Less than 2,5 Tn	Between 2,5 and 4 Tn	Between 4 and 6 Tn	Between 6 and 10 Tn	Between 10 and 15 Tn	Between 15 and 25 Tn	More than 40 Tn
Range velocity (m/s)	0,62	0,45	0,31	0,23	0,15	0,1	0,08

After that, observing every movement separately, though every acceleration must be applied in the corresponding direction:

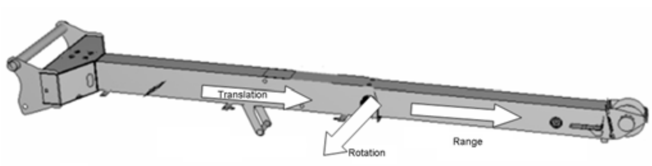

Figure 1: Movements in a telescopic crane jib

- **Translation:** it is the due to the translation of the derrick, in our case of the forklift truck, especially during the stopping process, where the maximum deceleration will take place. The calculation will be realized introducing acceleration in longitudinal direction to the crane jib. To know this acceleration, it would be suitable to know the minimal time of deceleration, that is unknown, so it is going to be estimated approximately for a second. Then, the maximum deceleration would be approximately of 2.8 m/s^2 that is a reasonable value.
- **Rotation:** it takes place only in gyratory fork-lift trucks, when it tends to turn and its maximum value will happen with the maximum scope of the machine with the crane jib. In this case there arise two types of tangential accelerations, on the one hand owed by centrifuge forces and on the other hand by acceleration or deceleration necessary to reach the speed of rotation or to stop, being this last case witch presents biggest decelerations. So:

$$a_{centrif} = \frac{v_{tra}^2}{r} \qquad (9)$$

and for the most unfavourable case is 2.2 m/s^2, although it would change according to the distance of every zone.

$$a_{acel\_fren,\max} = \frac{v_{rot,\max}}{t_{stopping}} - a_{centrif} = \frac{\Omega_{rot} \cdot r_{alc,\max}}{t_{stopping}} - \Omega_{rot}^2 \cdot r_{trange,\max} \qquad (10)$$

$$a_{acel\_rot,\max} = \frac{v_{rot,\max}}{t_{stopping}}$$

And with 2 seconds stop time, that is obtained when the maximum deceleration in the rotation is 3 m/s^2

- **Scope:** it is the due one to the movement of the telescopic jib on having spread out or on having bent, but it presents a few very small speeds, so the decelerations or accelerations will be ridiculous in comparison to the rest and to the gravity.

V. WIND LOADS

The calculation of these loads, since it was mentioned previously is calculated according to the norm UNE 53-113 that it establishes that the force that acts on the incidental face opposite to the wind in normal conditions of using (w_i) is:

$$P_{wind,ele} = P \cdot C_{f,element} \qquad (11)$$

- P_{wind}: pressure wind, in this case it will be used 125 N/m^2

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(table 1 of UNE 53-113)

- $C_{f,element}$: It depends on the form of the element on which the air affects and, in our case it will refer in all the zones of exhibition for closed box crane jibs (table 2 of the norm UNE 53-113). In case of lattices 1.7 will be used except if it is a tubular one (table 2 of the norm UNE 53-113).

To introduce the mentioned load, it will be applied in a transversal direction to the crane jib in the all the incidental section. About the forces due to wind in tempest (w_o), these crane jibs are not designed to be employed then, but if it is necessary, it would be calculated using a value of P of 800 N/m^2 for heights minor of 20 m or of 1100 N/m2 for others.

VI. WELD CALCULUS

Norm UNE-58356 specifies the procedure to calcule the welds of the crane jib, so it is suitable to study it. Then, the calculation is similar to the rest of the cases, but the only thing that changes is the safety coefficient that is applied to this zone and that is necessary to verify them with regard to four criteria:

- Combined stress limit (using Von Mises)
- Traction component
- Compression component
- Shear stress

Table 2: Security coefficient in the welds

Combined limit stress	Traction stress limit			Compression stress limit		Shear stress limit	
	Butt weld without preparation	Butt weld with preparation	Weld cord	Butt Weld	Weld cord	Butt Weld	Weld cord
1,11	1,11	1,25	1,57	1,11	1,39	1,57	1,92

VII. FEM APLICATION DEL IN A PARTICULAR CASE

The method of the finite elements (M.E.F.), presents huge advantages for the calculation and design of these elements, allowing an easy and a simple form to the modify the thicknesses and materials to obtain the stress and displacement maps[2,3,4]. Due to the simplicity that the calculation of lattice and closed box structures presents, there will be exposed the methodology used for a telescopic crane jib, which principles will be able to apply to the rest of the crane jibs directly. (See Fig. 2). For this particular case there has been used the commercial program ABAQUS to realize the FEM simulation.

Figure 2: FEM model for a telescopio crane jib

Material Model

To simulate a material, it has split of the curved stress - strain of each material and, supposing that we are always in the elastic zone, the only values that we need are the elastic limits and Young's module.

Welds Model

The method of the finite elements allows the perfect simulation of the welds (see figures 3 and 4). So, if a weld is observed thoroughly, it is possible to to see that, due to the process of welding some heat affected zones appear in the surroundings of the weld cord. These zones will be named as HAZ's and the zone that stays under the cord of weld as ZBS. After doing in the above mentioned zones a hardness test, there can be obtained the properties of the mentioned zones after the welding process, and usually they have an elastic major limit higher but they have worse fatigue properties (except for high elastic limit steels). Then, because they have a bigger elastic limit, the elastic normal limit is used for them them.

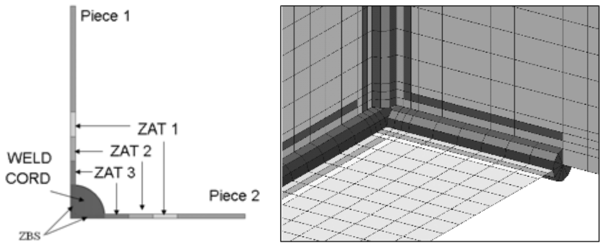

Fig. 3 y 4: Butt weld zone. Corner weld model with ZBS y HAZs

So to simulate the welds it is not necessary to realize a fatigue analysis of the weld, only it would be necessary to shape the weld cord using volumetric elements. On the other hand, the weld model depends on the form on the weld. In case of a butt weld the modelling would be realized like it can be appreciated in the figure 5, joining the diverse pieces in the corners and in the weld.

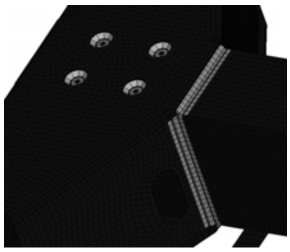

Fig. 5: Butt weld zone

Nevertheless, in lap welds, the simulation process is not so simple, because the unions take place in the zones of union, but in the interior it is not known certainly how the materials act and how they distribute the stress between them. Then for this type of union it is necessary to establish a different simulation model, although, for small lap welds it is possible to consider that in the interior have a hardness joint of both pieces.

In this particular case the weld zone appeared a lap weld, which is the joint between the box and the forklift truck connection, that was not possible to realize the simplification,

Since it can be seen in the figure 4, in the top zone a weld appears a lap weld with a few weld buttons in the centre. So, it was necessary to model this zone with volumetric elements to simulate perfectly the weld cords.

In this zone the diverse zones were joined using weld cords and, in the rest of the lap zone there is a contact with friction between the diverse pieces to join. With all this it is posible to model and calculate all the welds with the method of the finite elements.

Hydraulic cylinder model

Telescopic crane jibs, contain in its interior a hydraulic cylinder that allows to extend or to re-bring the jibs. This element, in its position of rest, so much extended as shrunk, only transmits traction/compression strains between both parts of the jib, so to simulate it, it has been replaced for a rigid bar which only can transmit traction/compression strains.

On the other hand, this element, along its process of extension, transmits some forces between diverse zones that allow the relative displacement of the parts. These forces are difficult to simulate, because they depend on the hydraulic system, but, is possible to model the accelerations that the front box has and they will have a maximum value similar to the acceleration that are taking place when the range increase. They were considered despicable so it is not necessary to include them.

Simulation of the not structural elements

The crane jibs have sometimes not structural elements that have a non-despicable mass: lines, valves, the own weight of the hydraulic cylinder with the oil, etc.

These masses are sometimes off-centred with the axis of the crane jib, so this generates a torsion that must be included. So, to simulte it, there are placed in the union zones of these elements with the crane jib forces and equivalent moments.

Simulation of the friction shoes

To allow the relative displacement of both box in the telescopic plumines and to allow the transmission of strains between them, the friction shoes are used. These elements are fixed to one of two bodies and prevent that the friction between the bodies will be excessive and could damage the machine.

Then it is necessary to include these pieces for a perfect design of the jib and to simulate the stress that appears in the structure, allowing veraciously model in stress and in displacement terms, because they allow a relative turn between both bodies.

So which is done it is to simulate these elements joined ton the body as volumetric elements and fitting the tolerances to the established ones during the manufacturing process. With regard to another body, some contacts with friction are established.

Load Cases

The load cases will be those ones that were established in the paragraph 4, and, for this particular case for the most unfavourable situation of the crane, which will take place when it is totally extended, becouse then, the strain in the diverse components will be maximum.

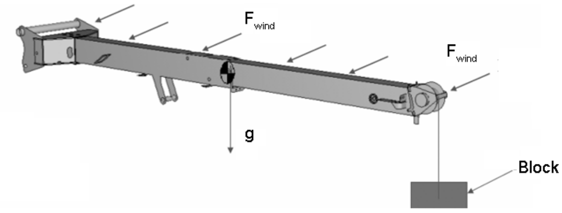

Fig. 6: Load Case 2 for a telescopio crane jib

For our particular crane jib, due to its modes of use, there have been included three additional load cases that are suitable to analyze. These cases are:

Additional case 1: crane jib with the loads of the load case 1 and with a 30º inclination angle of and dragging the maximum capacity of load.

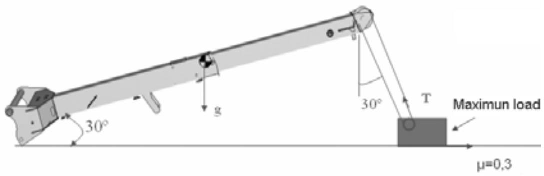

Fig. 7: Adicional case 1 for telescopio crane jib

This load case should not be produced along the life of the crane jib, because it is not been designed to work like this, but, it has been included asa additional safety calculation becouse this type of maneuvers occurs sometimes. To realize this case, the directions of the gravity would be modified in the finite elements model and the applied forces; the service load *(F)* would be replaced with the value of *T* and its application direction would be changed. The value of T is obtained of the following expression:

$$T = \frac{F \cdot \mu}{\cos(\alpha)} \qquad (12)$$

where μ is the friction coefficient of the block with the ground and it has been used a value of 0.3 and α the angle between the jib and the horizontal.

Adicional Case 2: crane jib with the loads of the load case 1 and dragging laterally the maximum capacity of load with a 30º. angle

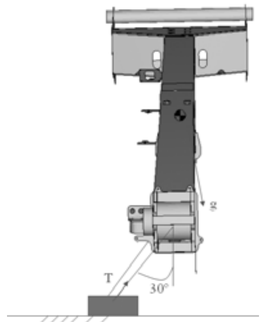

Fig. 8: Adicional case 2 for telescopio crane jib

This case is calculated similarly to the load case 1, but only the angle of application changes of *F* and value that is replaced with the value of the tension (*T*) that is obtained from the equation 12

Case 3: Case of Load 1 in middle scope.

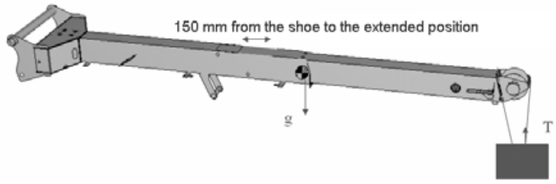

Fig. 9: Additional load case 3 for a telescopic crane jib.

The telescopic crane jib is designed to be employed only at two positions: totally spread and totally widespread, but there has been included this type of safety load case.

It is necessary to emphasize that these crane jibs in the contact zones for the totally spread position usually have local batten (over thickness or bears). So it is possible that in an intermediate load case of displacement of the boxes, these elements do not actuate and can appear local stress concentrations. Then, in this load case, it is defined an intermediate position, but more closed to the position of maximum spread.

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Obtained Results

There have been analyzed the tensions and displacements maps for the telescopic crane jib that has been calculated (see figures 10 and 11).

Fig. 10: Von Mises Stress for the additional load case 1

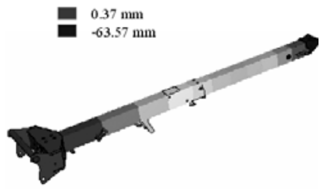

Fig.11: Vertical displacements for the adicional load case 3

After the design and optimization process, the obtained telescopic optimized crane jib keep the stress and strain criteria specifies by the regulation and by the company, with a final weight of 434.7 Kg. Opposite to the initial one of 498.2 Kg., that supposes a 12 % weigh saving.

To do the optimization there are used the limits established in the equations 6 to 8 for each one of the diverse pieces. So, in each of these pieces there has been examined Von Mises stress, the shear stress and the buckling. Later with each of the stress the safety coefficient is obtained using the following equation:

$$C.Seg = \frac{\sigma_{e\_mat}}{\max(\sigma_{V.M.Max}, \tau_{\max}, \sigma_{comp.\max})}$$

If C.Seg is bigger than the established one for the designer (in our particular case 2) it is possible the thickness reduction of the piece or using materials with worse properties. In case that it is lower it will be preceded increasing thicknesses or improving the quality of the used material. This will be realized with the important pieces, taking into account the available materials for the company, the thicknesses, etc. After that there will be realized again the calculation and will be analyzed to observe the behaviour of the modified pieces and to see if it fulfils the established criteria, being able to proceed successively up to coming to an ideal configuration of materials and thicknesses.

Then, iteratively it is possible to find a thickness and material optimized solution that fulfils the regulation and that has the lowest possible weight. It is possible to use also optimization programs like OPTIMUS that realizes this labour automatically.

VIII. ETENSIOMETRICAL ANALYSIS

To verify the behaviour of the mechanical set ant to validate the numerical calculation methodology and to verify the deviation of the numerical results with the reality, an extensiometrical analysis has been realized of the mechanical set, in which with three unidirectional gage (see figure 12) the deformations have been obtained in three previously selected points of the crane jib.

Fig. 12: Extensiometrial gage 1(left.) y 2(right.)

To realize the numerical - experimental correlation the load case that has been used is the load case 5. The other load cases are difficult to du experimentally because it is difficult to simulate wind loads. It is necessary to emphasize that with this load case the used methodology and the numerical results can be validated, so it is not necessary to realize other tests with other load cases.

Graph 1 shows that gage 1 and 3 suffer traction loads and gage 2 suffer compression loads.

As it can be observed in figures 13 to 15, the experimental and numerical results present a high degree of correlation, with a 12% maximum deviation that appears in gage 3 and it presents the lowest deformation.

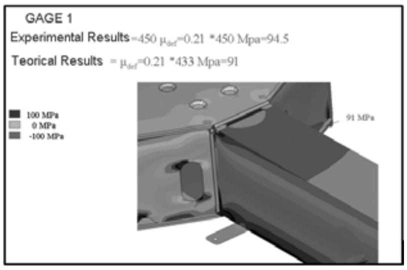

Fig. 13: Numerical – experimental analysis for gage 1

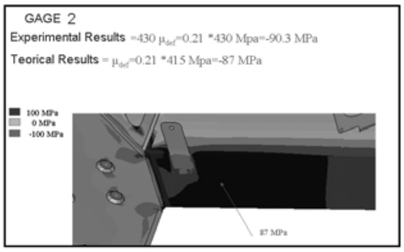

Fig. 14: Numerical – experimental analysis for gage 2

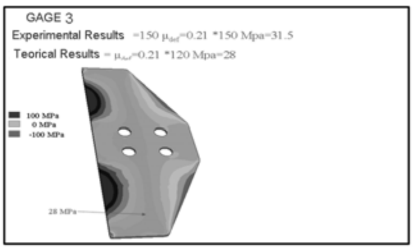

Fig. 15: Numerical – experimental analysis for gage 3

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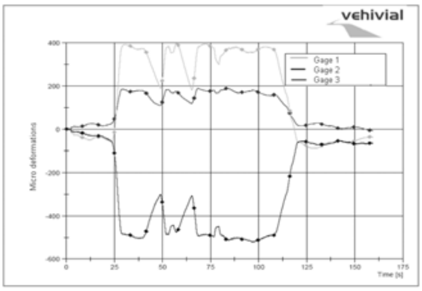

Graphic 1: micro deformations vs. time in the extensiometrical analysis for each gage

IX. CONCLUSIONS

The principal conclusions that are obtained are that it has been developed a methodology for the design, calculation and optimization of crane jibs for fork-lift trucks, adapting the regulation UNE-58536 for movile cranes, becouse the norm UNE-1726 is not too much restrictive; so it was necessary to define the load cases, the boundary conditions, the loads, the materials, the welds models, etc.

Furthermore some additional load cases not contemplated in the norm have been established and they have a great interest for a correct designo of the mechanical set, principally because the simulate some maneuvers that, although they are dissuaded or prohibited, can happen during the use of the crane jib.

Also a calculus methodology has been established using numerical technologies, especially finite elements method (F.E.M.), that allows to simulate perfectly all the elements and welds of the mechanical set and simulate all the load cases, and visualize in every point the stress, the strain, the displacements, etc.

On the other hand there has been checked the numerical results using extenosmetrical methods, obtaining a high degree of correlation and a low deviation (less than 12 %) that allows to validate the numerical results and the calculus methodology.

REFERENCE

[1] Sun, G.F; Kleeberger, M.; “Dynamic responses of hydraulic mobile crane with consideration of the drive system”. En “Mechanism And Machine Theory”. 4º edidion. Oxford: Ed. Edelweisser. 2003. Vol. 38, no12, pp. 1489-1508.

[2] Ortiz Berrocal L.; “Resistencia de materiales”. 3 Edition. Escuela Técnica Superior de Ingenieros Industriales: Madrid, 1980, pp. 112-187.

[3] Timoshenko, S.P.; ”Theory of elastic stability”. 2 Editio. New York: Mc Graw- Hill, 1961. pp 150-170.

[4] Zienkiewicz O.C. y Taylor R.L., “El Método de los Elementos Finitos. Vol. 1, las bases”. 8º Edition. Dover, Ed. Dover 2006. pp. 134-179.