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OPTIMUM DESIGN OF OVERHEAD CRANE GIRDERS UNDER MOVING LOADS

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Abstract. *A design of crane bridge girders is formulated with inequality constraints as a minimum weight problem and a software program is prepared. The welded box type girders are modelled as a simply supported beam. Stresses and displacements are obtained while trolley with the loads are moving on the girder in maximum deflection position. The minimization of objective function is taken as minimization of the girder weight. The stresses, displacements, natural frequencies and sizes are restricted. In such a way that they will not exceed in the safe values defined by the standards which are DIN and FEM. The nonlinear problem is solved by means of practical design optimization method which is used in designing huge mechanical systems. Numerical examples are given with illustrations which show the effectiveness of this approach.*

1 INTRODUCTION

Overhead cranes are basic facilities that are used to move heavy and high volume materials in many factories and plants. The fundamental motions of an overhead crane can be described as: object hoisting and lowering, trolley travel and bridge traverse. Overhead cranes with traveling bridge girders that are driven by electricity have a wide variety of applications. Since the main girders are the structures that sustain most of the load, careful attention has to be devoted to their safe and economical design. In this study, all formulations of double girder overhead cranes with box section profile are made on the basis of the geometrical measures of the box section (B, h, t_1, t_2) and these values are determined as design parameters. Design parameters have a direct effect on weight per unit length and total girder weight. Hoisting load, span and trolley weight are the important design data for a crane. The restrictions are imposed on the computer software which aimed at checking whether values for stress, deflection, buckling, vibration and geometric sizes conform to the safe values determined by standards. The lowest weight that complies with all these constraints is assumed to be optimal weight. In optimal design of overhead crane girders, considering the multitude of data, it is difficult to implement standard optimization methods. Therefore, it is more sensible to apply a practical design optimization method that is preferred in designing large scale mechanical system. Various studies have been carried out on overhead cranes. S.W. Cho and B.M. Kwak formulated the problem of the design of box-type bridge girders as a minimum weight design problem with inequality constraints [1]. K.A.F. Moustafa and A. Ebeid derived a nonlinear dynamical model for an overhead crane [2]. A.Z. Al Garni and his colleagues considered a nonlinear dynamic model of an overhead crane which represents simultaneous travel, traverse and hoisting/lowering motions [3]. M. Şimşek investigated vibration of a functionally graded simply supported beam under a moving mass by using different beam theories [4]. T. Niezgodzinsky and T. Kubiak analysed the local loss of stability in box girders of overhead cranes [5].

2 STRUCTURAL ANALYSIS OF DOUBLE-GIRDER CRANE WİTH BOX-SECTION

A box girder model is illustrated in Fig.1. First, the centre of gravity of the system is determined and then inertia moments I_x and I_y are calculated by means of the Steiner Theorem. The general formula for the weight is,

$$G = \rho\, A\, L \quad \text{(kg)} \qquad (1)$$

Where, A is the area of the box section, ρ is the density of material and L is the girder length. Rail and stiffener weights are included to the total weight of the girder. q is unit weight.

$$q = G/L \quad \text{(kg/m)} \qquad (2)$$

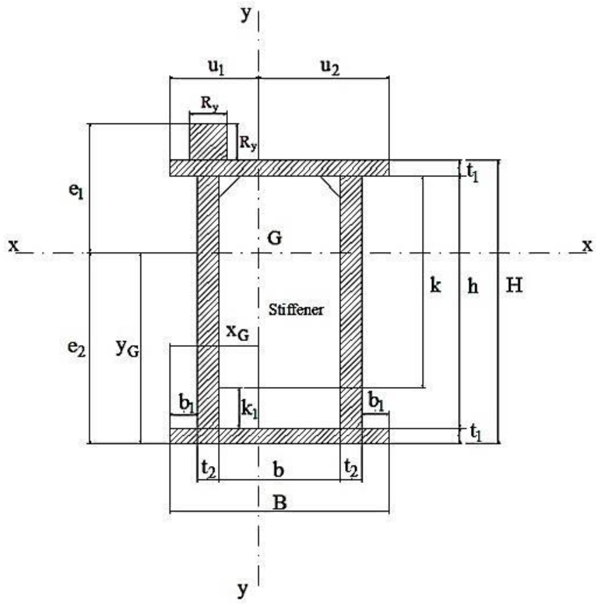

Figure 1: A box girder section.

2.1 Vertical loads

A girder model for design analysis is illustrated in Fig.2.

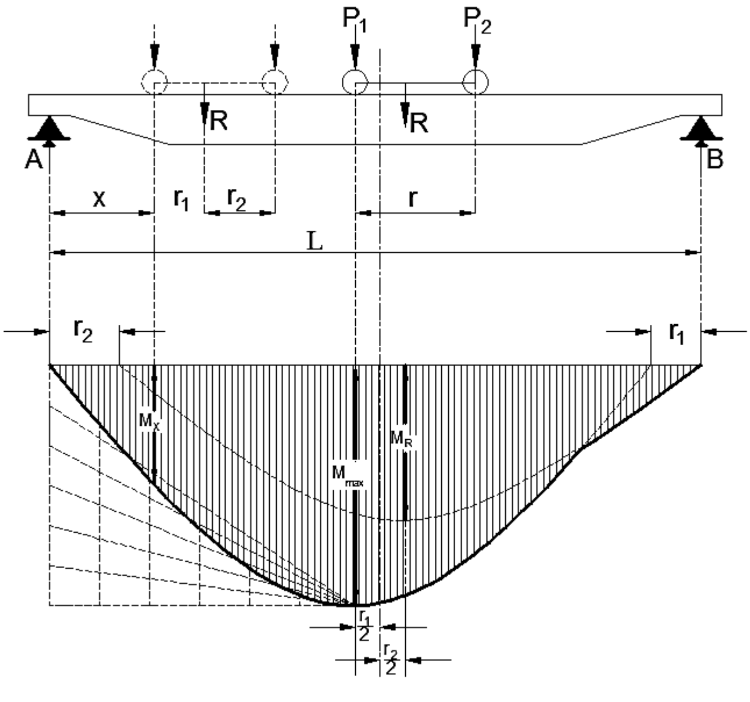

Figure 2: Moving loads on the girder and bending moments [12].

The girder is assumed to be a simply supported beam of a uniform cross section. The girder structure is assumed to be constructed with the same material. Stresses and deflections are obtained for critical conditions when all loads act simultaneously. Uniformly distributed load is the dead weight of the girder. The weights of the side walk on the girder are included in the uniformly distributed load. The moving loads are hoisting load P_L and weight of trolley P_T.

The whole loads are distributed across two girders. The concentrated load P is assumed to act on the two points at the wheel base of the trolley for each girder. Analysis is made for the case in which the trolley is the maximum deflection point of the girder.

Maximum bending stress σ_1 is induced by distributed load (q.L)

$$\sigma_1=\frac{q.L^2}{8.W_x} \quad (N/mm^2) \tag{3}$$

$$W_x= I_x / e_{max} \tag{4}$$

Gravity acceleration (g) is included in the formula.

The bending moment caused by moving loads can be investigated on the basis of the following equation.

$$M_x=F_A.x=\frac{R}{L}\ (L-x-r_1).x \tag{5}$$

F_A is a force on the A support in Fig.2.

$$x=\frac{L}{2}-\frac{r_1}{2} \tag{6}$$

x is obtained from $\frac{dM_x}{d_x}=0$ (7)

Where, x is the distance from the left girder support to the left trolley wheel. The maximum bending moment which takes place due to the left wheel load can occur when the left wheel is on the left hand side of the middle of the girder with the distance of $r_1/2$. The similar bending moment can obtained for the right wheel.

$$P_1+P_2=R \quad \text{and} \quad r_1+r_2=r \tag{8}$$

Maximum moment for the left wheel is calculated using the formula.

$$M_{max}=\frac{R}{4L}\ (L-r_1)^2 \tag{9}$$

If $P_1=P_2=P=\frac{R}{2}$ and $P_1+P_2=2P=R$ and $r_1=r_2=\frac{r}{2}$ (10)

Vertical bending moment caused by the weight of the trolley (P_T) and stress is,

$$\sigma_2=\frac{P_T}{32L\ W_x}\ (2L-r)^2 \quad (N/mm^2) \tag{11}$$

Vertical bending moment caused by the weight of the load (P_L) and stress is,

$$\sigma_3=\frac{P_L}{32L\ .W_x}\ (2L-r)^2 \quad (N/mm^2) \tag{12}$$

2.2 Horizontal forces

In this study, the horizontal forces and moments were calculated according to DIN 15018. Horizontal forces are inertia forces induced by the acceleration or deceleration of the girder and trolley, as well as wind loads. σ_4 is stress for the girder and σ_5 is stress for the trolley .

2.3 Total Normal Stress

$$\sigma_{max}=M(\sigma_{max1}+\sigma_{max2}+\Psi\,\sigma_{max3}+\sigma_{max4}+\sigma_{max5}) \quad (\mathrm{N/mm^2}) \tag{13}$$

M is the coefficient of load group and Ψ is the vibration coefficient of the hoisting load.

2.4 Shear stress τ

In this study the box girder is asymmetrical. According to the torsion center, torsional moment τ_t, determined from vertical and horizontal forces on the trolley wheels. Because of bending stress along the span of the girder, shear flow is generated that results in the shear stress τ_s .

The total maximum shear stress τ_{max} is,

$$\tau_{max}=\tau_t+\tau_s \tag{14}$$

2.5 Combined stress σ_v

Since the bending stress and shear stress act simultaneously, combined stress σ_v is,

$$\sigma_v=\sqrt{\sigma_{max}{}^2+3\tau_{max}{}^2} \quad (\mathrm{N/mm^2}) \tag{15}$$

2.6 Deflection

Total deflection of the girder is obtained by superposing two deflections .

Maximum deflection f_1 due to the distributed load (P_d=q.L) is,

$$f_1=\frac{5P_dL^3}{384\,E\,I_x} \quad (\mathrm{mm}) \tag{16}$$

Maximum deflection f_2 due to concentrated load P_1 is,

$$f_2=\frac{P_1}{48\,E\,I_x}(r^3+2L^3-3Lr^2) \quad (\mathrm{mm}) \tag{17}$$

E ($\mathrm{N/mm^2}$) is material elasticity module

$$P_1=(P_A+\Psi P_Y)/4 \tag{18}$$

Total deflection f is,

$$f=f_1+f_2 \tag{19}$$

2.7 Vibration analysis with Mohrsche Procedure

This method can be applied to overhead cranes. The mathematical model of the method is taken from the literature [13]. Only a single concentrated mass is present, the moment of the system is only determined by the vertical shift U in Figure 3.

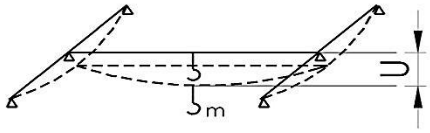

Figure 3: Static deflection of the system [13].

$$U = \frac{G}{k} = \frac{mg}{k} \tag{20}$$

Static deflection $U = U_G + U_P$

Whereas U_G is dependent on the starting conditions, U_P depends on the forcing P.

$$W_0 = \sqrt{\frac{k}{m}} = \sqrt{\frac{g}{U}} = \frac{31{,}25}{\sqrt{U}} \qquad (s^{-1}) \tag{21}$$

$$f_0 = \frac{W_0}{2\pi} = \frac{5}{\sqrt{U}} \qquad \text{(Hz)} \tag{22}$$

Natural frequency f_0 is,

$$f_0 = \frac{5}{\sqrt{U}} = \frac{5}{\sqrt{\frac{PL^3}{48EI}.\eta}} = 34.6\sqrt{\frac{EI}{PL^3}}\eta \qquad \text{[Hz]} \tag{23}$$

η was calculated.

3 FORMULATION OF OPTIMAL DESIGN PROBLEM

The objective function is expressed in the following formula from Fig.1.

$$G = L\,\rho\,(2t_1B + 2t_2h + R_Y^2) \tag{24}$$

The design parameters are expressed as follows.

$$\emptyset = (b, h, t_1, t_2) \tag{25}$$

The constraints are taken as inequality constraints.

1. Constraint: Combined stress control,

$$\emptyset_1 = \sqrt{\sigma_{max}{}^2 + 3\tau_{max}{}^2} - \sigma_a \leq 0 \tag{26}$$

σ_a is a allowable stress. It was taken from the specifications for the material selected.

2. Constraint: Deflection control,

$$\emptyset_2 = f_1 + f_2 - f_a \leq 0 \tag{27}$$

f_a is allowable deflection. It is taken as L/f values in range of 1000-1500 in this study.

3. Constraint: Control of surface buckling in the lateral web plates by DIN 4114,

$$\emptyset_3 = (\min \nu_B - \sigma_{VK}/\sigma_V) \leq 0 \tag{28}$$

4. Constraint: Control of surface buckling in the upper flange plates by DIN 4114,

$$\emptyset_4 = (\min \nu_B - \sigma_{VP}/\sigma_V) \leq 0 \tag{29}$$

5. Constraint: Vibration frequency control,

$$\emptyset_5 = f_0 - f_{0a} \leq 0 \tag{30}$$

where, f_0 denotes natural frequency of the girder. f_{0a} was employed as the limit value making sure that the maximum deflection did not exceed allowable deflection value.

Geometrical constraints, B=290-690 mm , h=490-1490 mm , t_1=6-12 mm , t_2=6-10 mm

4 SOLUTION PROCEDURE

The numerical examples were solved by means of the computer software given in fig.4.

mm	B	h	b	H	t_1	t_2	R_v
K01	290	490	218	502	6	6	40
K02	290	490	218	506	8	6	40
K03	290	690	218	702	6	6	40
K04	290	690	218	706	8	6	40
K05	290	690	218	710	10	6	40
K06	490	690	418	702	6	6	40
K07	490	690	418	706	8	6	40
K08	490	690	418	710	10	6	40
K09	490	690	418	710	10	8	40
K10	490	990	418	1002	6	6	40
K11	490	990	418	1006	8	6	40
K12	490	990	418	1010	10	6	40

Table 1: Box-girder sizes

Example

Design Variables		**Selected Variables by [9]**	
Hoisted load	P_L =16 ton	Hoist class	: H2
Span	L =15 m	Loading Group	: B3
Trolley weight	P_T = 1700 kg	Crane load coefficient	: M
Trolley wheel base	r = 1.2 m	Notch effect coefficient	: K3
Deflection ratio	L/f=1000	Own weight coefficient	: φ=1.1
Hoisting speed	v_H=5 (m/min)	Material of girder	: St37
Traveling speed for crane	v_C=25 (m/min)	Elasticity module	E = $2{,}1.10^{11}$ (N/m^2)
Traveling speed for trolley	v_T=15 (m/min)	Density of material	ρ=7,85 (gr/cm^3)

OPTIMUM BOX-GIRDER K07
h/B= 1.4

PARAMETERS

B (mm) =490
h (mm) =690
tl (mm) = 8
t2 (mm) = 6

CALCULATIONS

B (mm) =418
H (mm) =706
A (mm^2) = 17720
q (Kg/m) = 149.76
G (Kg) =2249.46

1. Constraint = -84.026
2. Constraint = -0.913
3. Constraint = -1.651
4. Constraint = -1.173
5. Constraint = -3.741

Inertia moment by X-X axis	Ix (mm^4)	= 6.686928E+08
Inertia moment by Y-Y axis	Iy (mm^4)	= 1.48619E+09
Stress 1 by the distributed load	σ1 (N/mm^2)	= 13.886
Stress 2 by the weight of the girder	σ2 (N/mm^2)	= 7.643
Stress 3 by the weight of the trolley	σ3 (N/mm^2)	= 26.976
Stress 4 by the horizontal inertia forces	σ4 (N/mm^2)	= 19.456
Stress 5 by the horizontal forces for trolley	σ5 (N/mm^2)	= 2.007
Maximum normal stress	σmax (N/mm^2)	= 75.956
Maximum shear stress	τ (N/mm^2)	= 0.961
Combined stress	σ_v (N/mm^2)	= 75.974
Surface buckling for lateral plate	σ_{vk} (N/mm^2)	= 228.000
Surface buckling for upper plate	σ_{vp} (N/mm^2)	= 219.000
Maximum deflection	f(mm)	= 14.087
Naturel frequency	f_0 (Hz)	= 2.631

Figure 4: The computer printout

5 RESULTS

The design parameters in Table 1 were tested in order starting from K01 and the solution fulfilling all constraints was obtained with the dimensions of girder K07. For the same example h/B cross sectional ratios were varied and new design variables were obtained. The B values of 290, 490 and 690 mm in this example were taken from Table 1. The h/B ratios were tested by means of a software program and the ratios that fulfill all constraints were determined. For this example, the optimal region was established as follows.

B=290 for h/B=2,8 - 3 and B=490 for h/B=1,4 - 3 and B=690 for h/B=1,4 - 2,2

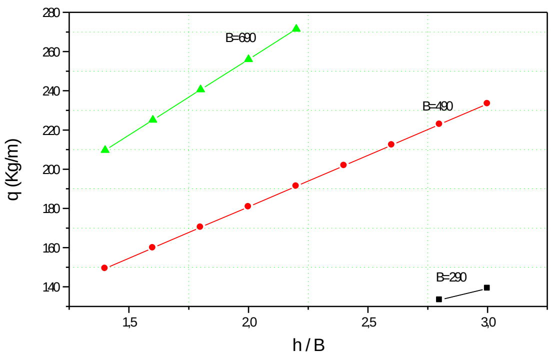

Figure 5: Unit weight of girder – h/B

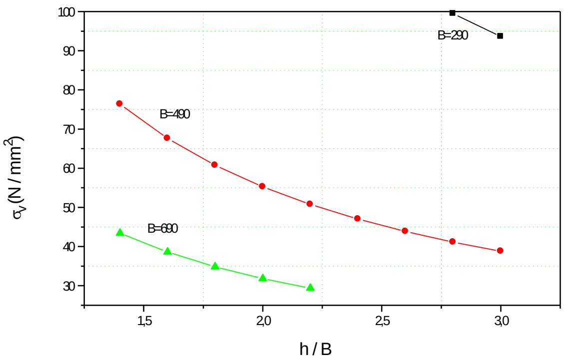

Figure 6: Combined stress of girder – h/B

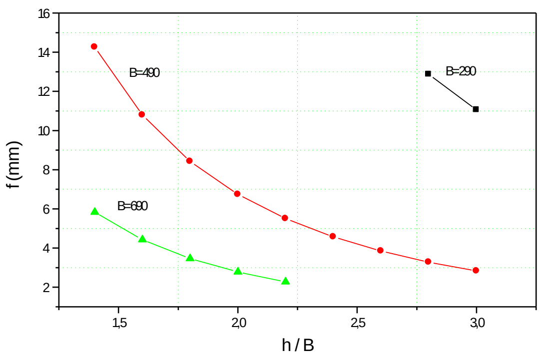

Figure 7: Deflection of girder – h/B

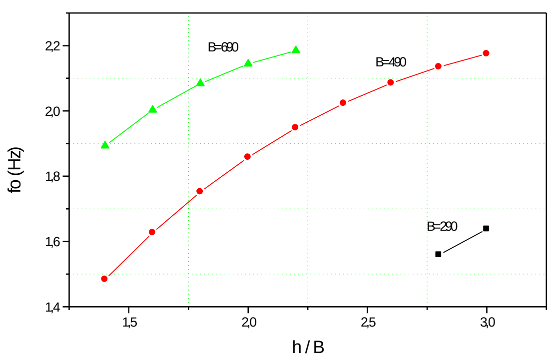

Figure 8: Natural frequency of girder – h/B

When any parameter reduction is done, where one or more constraints approach to limit values, it was assumed that a local optimum value was achieved. The optimal region was determined by the local optimum values. It is illustrated optimum combined stress values in Fig.6, optimum deflection values in Fig.7, optimum naturel frequency values in Fig 8. As is evident from Fig.1, it was only if h/B=2,8 for B=290 mm that all constraints were fulfilled. The value of q=133,27 kg/m is the minimum unit weight in the optimal region. For B=490 mm all the constraints were satisfied only if h/B=1,4 and value of q=149,34 kg/m in this group is the minimum unit weight. B=690 mm is too much safety for this example. Even h/B values less than 1 fulfill the constraints and these h/B ratios are not appropriate. A geometrical form of h/B≤1 is not desirable. Moreover, if h/B≥2,2, h value exceeds 1500 mm. This is not desirable conditions. Increasing the cross sectional ratio too much will bring about instability problems. When reducing the values, priority was given to lateral plate and the attempt was made to determine the thinnest lateral plate that fulfilled the constraints. I tried to obtain the lowest values that fulfilled the constraints. Deflection is the most frequently violated constraint during the optimization. The section ratio reached the limit value in long span cranes. Combined stress constraints are highly active because the girder is designed a torsional box type. Surface buckling constraints became active in the lateral and upper plates.

6 CONCLUSIONS

The optimal design results were compared with those of girders that have been developed and produced by heavy industries in recent times. An attempt was made to minimize girder weight as an optimization criterion and particular importance was attached to the issue of safety at every stage of the problem. The optimization method developed to this end produced a much more sensitive study. Apart from materials economy thanks to reduction of girder weight, the sustaining structure is also made lighter.

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