Optimalno projektovanje zavarenog I-nosača strele stubne konzolne dizalice

Goran Pavlović^{1*}, Mile Savković², Goran Marković², Nebojša Zdravković², Jelena Stanojković¹

¹Istraživačko-razvojni centar Alfatec d.o.o., Niš (Srbija)

²Fakultet za mašinstvo i građevinarstvo u Kraljevu, Univerziteta u Kragujevcu, Kraljevo (Srbija)

U ovom radu predstavljen je postupak analize i optimizacije konstrukcije strele stubne konzolne dizalice. Smanjenje mase zavarenog I-nosača strele, odnosno površine poprečnog preseka nosača, postavljeno je kao glavni cilj ovog istraživanja, pri čemu je suština da se standardni valjani I-profil zameni zavarenim. Kao glavne funkcije ograničenja procesa optimizacije, pored kriterijuma dozvoljenog ugiba i dozvoljenog napona, posmatrani su i naponi u šavovima, kako u samom nosaču, tako i u vezi nosača sa okretnim delom konstrukcije. Takođe, korišćene su i određene konstruktivne i tehnološke preporuke, koje predstavljaju dodatne funkcije ograničenja u postupku optimizacije. Optimizacija je izvršena korišćenjem MATLAB softverskog paketa, primenom algoritma ubrzanog roja čestica (APSO), algoritma svitca (FA) i algoritma kukavičje pretrage (CSA), kao i Ms EXCEL softverskog paketa, korišćenjem alata Solver, gde su upotrebljeni algoritmi optimizacije za nelinearne probleme sa ograničenjem (metod generalizovanog redukovano gradijenta i evolucioni algoritam optimizacije). Dobijeni rezultati verifikovani su na realnim primerima konstrukcija stubnih konzolnih dizalica i pokazali opravdanost primene datog postupka.

Ključne reči: Stubna konzolna dizalica, Matlab, Ms Excel, Optimizacija, Zavareni nosač

1. UVOD

Današnja industrija zahteva svestranu, efikasnu i ekonomičnu opremu, a istovremeno treba da pruža više fleksibilnosti uz značajne uštede kroz povećanu produktivnost. Stubne konzolne dizalice mogu znatno pomoći u poboljšanju efikasnosti manipulacije materijalom i tokovima rada i proizvodnje. Ozbiljnija razmatranje treba da se odnose kod primene ovih dizalica koje zahtevaju češća ponovljanja podizanje i prenos tereta unutar fiksnog luka rotacije dizalice.

Potreba kontinualnog poboljšanja u tehnologijama manipulacije materijalom je zajednička karakteristika mnogih modernih inženjerskih problema i poduhvata.

Pravilan izbor opreme i dimenzionisanje segmenata celokupnog sistema je od velikog značaja. Potrebno je da budu ispunjeni svi neophodni uslovi kako bi oprema bila stabilna, sigurna i bezbedan za rad, a takođe treba voditi računa i o ceni koštanja. Zato je neophodna detaljna analiza i optimizacija parametara konstrukcije dizalice. Treba da se obezbedi smanjenje broja servisiranja, kao i nesmetano funkcionisanje opreme bez brekida u radu.

Konzolni deo celokupne strukture stubne dizalice je najodgovorniji deo strukture, i predmet je analize i istraživanja u ovom radu. Konstrukcija stuba nije tema analize i optimizacije, pošto se radi o segmentu izrađenom od standardne šavne cevi, tako da optimizacija ove geometrije nije od značaja za primenjeni postupak optimizacije u ovom istraživanju, zbog samog ekonomskog aspekta, što se tiče fabrikovanja ovakvih tipova nosača.

Optimizacija predstavlja proces gde se dobijaju najbolje vrednosti parametara optimizacije (varijabli) na osnovu zadatih funkcija ograničenja, kao i donjih i gornjih graničnih vrednosti parametara, za posmatranu funkciju cilja. Najzastupljeniji su postupci optimizacije koje imaju za cilj minimizaciju mase, cene koštanja i slično. U zadnje

vreme su sve više prisutne multi-objektivne optimizacije, gde postoji više funkcija cilja u procesu optimizacije.

Iz navedenih razloga i zbog značaja ovih tipova konstrukcija, prisutan je veliki broj istraživanja i publikacija koje se bave problemima analize i optimizacije struktura ovih tipova dizalica, a naročita konstrukcije strele konzolne dizalice, kao njenog najvažnijeg i najodgovornijeg segmenta.

Zbog samog načina izvođenja i oblika struktura konzolnih dizalica, kao najznačajniji kriterijum i uslov u procesu analize i optimizacije se uzima ugib vrha strele konzole, a takođe se posmatraju i naponska stanja cele strukture.

Analiza ugiba i naponskih stanja se u najvećem broju slučajeva vrši primenom MKE, pri čemu se rezultati porede sa analitičkim ([1], [2], [3], [4], [6], [10] i [11]). U radu [1] izvršena je analiza ugiba i naponskih stanja strele konzolne dizalice primenom ANSYS softverskog paketa, pri čemu je 3D model izveden u CATIA softverskom paketu. Pored tipičnog standardnog I-profila koji se najčešće koristi kod ovih konstrukcija, posmatran je i standardni kutijasti profil. Slično prethodnom, u radu [2] je primenom COSMOS softverskog paketa izvršena analiza I-profila strele, dok je 3D model generisan u SOLIDWORKS softverskom paketu.

U radovima [3] i [4], za razliku od prethodno navedenih radova, analizirane su celokupne strukture konzolnih dizalica (stub i konzolni deo). U radu [3] analizirani su različiti tipovi čelika na primeru jedne stubne konzolne dizalice primenom ANSYS softverskog paketa, dok je 3D model izrađen u softverskom paketu Pro/E. Date su preporuke za izbor tipa čelika, s obzirom na uslove dizalice u eksploataciji. U radu [4] autori su izvršili sveobuhvatnu analizu strukture jedne stubne konzolne dizalice, za različite položaje tereta na streli i ugla strele, dge je dopušteni ugib postavljen kao primarni kriterijum, pri čemu su rezultati verifikovani i eksperimentalnim

putem. MKE analiza je izvršena korišćenjem softverskih paketa Krasta i ANSYS. Ugib vrha strele stubne konzolne dizalice je dataljno analiziran u radu [5], za različite modele proračuna, gde su rezultati verifikovani primenom softverskih paketa CATIA i SAP2000.

Pored navedenih softverskih paketa za MKE, koji su najviše u upotrebi, primenu ima i NX 10.0, koji je primenjen u radu [6], gde je izvršena detaljna analiza konstrukcije stuba jedne konzolne dizalice. Primenom softverskog paketa Inventor, u radu [7] izvršena je optimizacija strukture jedne zidne konzolne dizalice, rešetkaste strukture, u cilju smanjenja mase postojeće strukture.

Značaj pravilnog modeliranja i analize strukture primenom MKE, pokazan je u [8], gde je detaljno data procedura pravilnog modeliranja sklopa jedne stubne konzolne dizalice u softverskom paketu ANSYS u cilju pravilnog definisanja odnosa između segmenata strukture, kako bi se dobili što precizniji rezultati u analizi.

U prethodnim radovima su posmatrani u najvećem broju slučajeva standardni I-profil. U radu [9] je primenom softverskog paketa ANSYS analiziran promenljivi (konusni) I-poprečni presek, za različite vrednost veličine ugla konusa, uzimajući u obzir pored ugiba i napona, i elastičnu stabilnost nosača.

Pored ugiba i naponskih stanja, pojedini autori ukazuju i na značaj modalne analize strukture konzolne dizalice ([10], [11] i [12]). U radu [10] je primenom softverskog paketa ANSYS izvršena analiza jedne stubne konzolne dizalice, pri čemu je prikazana zavisnost frekvencija na napone i ugibe u određenim pravcima. U radu [11] je optimalan tip I-profila određen tako što je posmatrano nekoliko odabranih tipova standardnih I-profila, na osnovu analitičkih proračuna i MKE analiza, pri čemu su rezultati verifikovani i eksperimentalno. U istraživanju [12] autori prikazuju jedan od mogućih metoda modeliranja dinamike podizanja terete i njenog uticaja na I-profil strele stubne konzolne dizalice, pri čemu je simulacija dinamike podizanja tereta izvršena korišćenjem MATLAB-Simulink softvera.

Razvojem 3D softverskih paketa, u novije vreme se sve više primenjuju topološke analize i optimizacije, a njihov značaj i primena su prikazane u preglednom radu [13], kod I-profila i duplih I-profila. Takođe, u radu [14], na I-profilu konstrukcije varirani su različiti oblici strukture rebra profila i njegovi parametri, i praćeni su uticaj na ugibe i naponska stanja.

Za razliku od gorenavedenih publikacija, gde je primenjivana MKE, u radovima [15] i [16] je izvršena optimizacija poprečnih preseka konstrukcija strele korišćenjem određenih numeričkih algoritama optimizacije. U radu [15] je izvršena optimizacija sandučastog poprečnog preseka konstrukcije strele primenom evolucionog algoritma optimizacije (EA) u Ms EXCEL softverskom paketu i na taj način je ostvarena znatna ušteda u poređenju sa postojećim rešenjem. Slično prethodnom, korišćenjem poboljšanog genetskog algoritma optimizacije (IGA), u radu [16] optimizovani su geometrijski parametri konzolnog dela strukture dizalice, gde je na jednom primeru u višekriterijumskoj optimizaciji ostvarene ušteda od oko 20 %. Takođe, Ms EXCEL softverski paket je uspešno korišćen kod optimizacije

dizaličnih struktura jednogrednih mosnih dizalica, što je prikazano u radu [17].

U radu [18] je izvršena optimizacija nosača strukture dvogredne mosne dizalice, pri čemu je korišćen analitički metod optimizacije, i dobijeni su eksplicitne vrednost izraza za optimalne geometrijske parametre i njihove odnose.

Konačno, imajući u vidu prethodno navedene publikacije i značaj analize i optimizacije ovih tipova struktura, glavni cilj ovog istraživanja je analiza i višekriterijumska optimizacija geometrijskih parametara zavarenog I-nosača strele konzolne stuble dizalice. Analiza i optimizacija biće sprovedene na primerima konzolnih dizalica koje su u eksploataciji.

2. PRIMENJENI ALGORITMI OPTIMIZACIJE

U ovom istraživanju su primenjeni različiti numerički metodi (algoritmi) optimizacije, korišćenjem softverskih paketa MATLAB i Ms EXCEL. Primenjeni su metaheuristički algoritmi optimizacije, algoritam ubrzanog roja čestica (APSO), algoritam svitca (FA) i algoritam kukavičje pretrage (CSA) u MATLAB softverskom paketu. Takođe, korišćeni su i algoritam generalizovanog redukovano gradijenta (GRG2) i evolucionog algoritma (EA) u Ms EXCEL softverskom paketu.

Algoritam ubrzani roj čestica, algoritam svitca i algoritam kukavičje pretrage su prirodom inspirisani algoritmi optimizacije. Algoritam ubrzani roj čestica (APSO) i algoritam svitca (FA) su razvijeni 2008. godine od strane X. S. Yang-a. Isti autor je 2009. godine zajedno sa S. Deb predstavio algoritam kukavičje pretrage (CSA). Sva ova tri algoritma su u ovom istraživanju primenjena u svom izvornom kodu, [19], [20] i [21], respektivno.

Ms EXCEL Solver koristi metod generalisanog redukovano gradijenta (GRG2 algoritam) za optimizaciju nelinearnih problema i razvijen je od strane L. Lasdon-a i A. Varen-a, 2011. godine. Takođe, Solver koristi i evolucioni algoritam (EA) za rešavanje nelinearnih problema optimizacije.

3. MATEMATIČKA FORMULACIJA PROBLEMA OPTIMIZACIJE

Postupak optimizacije za ovaj inženjerski problem ima za cilj određivanje optimalnih geometrijskih parametara zavarenog I-nosača konstrukcije strele, koji će dovesti do smanjenja njegove mase, odnosno površine poprečnog preseka profila.

Problem optimizacije je definisan na sledeći način: minimizacija funkcije cilja

$$f(X) \quad (1)$$

u odnosu na funkcije ograničenja:

$$g_i(X) \leq 0 \quad (2)$$

Takođe, mora biti ispunjeno i sledeće:

$$l_i \leq X_i \leq u_i \quad (3)$$

gde su:

$f(X)$ - funkcija cilja,

$g_i(X) \leq 0, i = 1, \dots, m$ - funkcije ograničenja,

l_j, u_j - donja, odnosno gornja granica varijable,

$i = 1, \dots, m$ - broj funkcija ograničenja,

$j = 1, \dots, n$ - broj projektnih varijabli,

X - projektni vektor koga čine n projektnih promenljivih.

Projektno promenljive (varijable) su veličine koje treba odrediti tokom optimizacionog postupka (svaka projektna promenljiva je definisana svojom donjom i gornjom granicom).

Da bi se definisala funkcija cilja i funkcije ograničenja, najpre će biti predstavljen inženjerski problem koji je tema ovog istraživanja.

U ovom radu glavni predmeti istraživanja se odnose na analizu i optimizaciju I-profila konstrukcije strele stubne konzolne dizalice (Slika 1). Glavna ideja je da se detaljno analizira i optimizuje zavareni I-nosač, odnosno da se prikaže opravdanost izrade ovakvog nosača koji bi zamenio standardne INP ili IPE nosače koji se najčešće koriste kod ovih tipova konstrukcija, u cilju smanjenja mase nosača.

Na Slici 1 prikazana je jedan tip stubne konzolne dizalice i osnovni ulazni geometrijski parametri. Struktura se sastoji iz dva osnovna dela, konstrukcije stuba, visine H_s i prečnika D_k , i konstrukcije strele, dužine L_k , koja oko ose stuba rotira preko osovine, dužine H_l , na udaljenosti a od ose stuba. Vitlo se uzima u analizi na kraju strele, odnosno na udaljenosti L od ose stuba (najnepovoljniji položaj).

Slika 1: Stubna konzolna dizalica

Ulazne veličine koje su neophodne za ovu analizu su prikazane preko vektora ulaznih parametara:

$$\vec{x} = (Q, m_t, L, a, H_s, H_l, I_{x,S}, R_e, K_f \dots) \quad (4)$$

gde su:

Q - nosivost dizalice,

m_t - težina vitla,

L - dužina luka rotacije dizalice,

$I_{x,S}$ - glavni moment inercije poprečnog preseka stuba,

R_e - napon granice tečenja materijala,

K_f - koeficijent koji zavisi od načina upravljanja dizalicom i pogonske klase, prema [22].

Vektor parametara optimizacije (projektnih promenljivih) glasi:

$$X = (x_1 \ x_2 \ x_3 \ x_4 \ x_5)^T = (b \ t \ h \ s \ a_s)^T \quad (5)$$

U daljem tekstu će biti prikazani detaljno parametri optimizacije (varijable), funkcija cilja i funkcije ograničenja.

4. FUNKCIJA CILJA I FUNKCIJE OGRANIČENJA

4.1. Funkcija cilja

Funkcija cilja predstavlja površinu I-poprečnog preseka zavarenog nosača strele konzole (Slika 2).

Slika 2: I-poprečni presek zavarenog nosača strele konzole

Površina poprečnog preseka zavarenog nosača, odnosno funkcija cilja, glasi:

$$A = 2 \cdot b \cdot t + h \cdot s \quad (6)$$

gde su:

b - širina profila zavarenog nosača,

t - debljina pojasnog lima profila zavarenog nosača,

h - visina rebra profila zavarenog nosača,

s - debljina rebra profila zavarenog nosača.

Pomenute veličine predstavljaju projektno promenljive u procesu optimizacije.

Debljina šava a_s ne figuriše u funkciji cilja, njen uticaj je indirektan na optimalne parametre poprečnog preseka I-profila, preko geometrijskih karakteristika koji utiču u kriterijumima gde je prisutan napon u šavu. Ista debljina šava je usvojena za oba pomenuta kriterijuma.

Matematička formulacija funkcije cilja prikazana je na sledeći način:

$$f(X) = A(x_1 \ x_2 \ x_3 \ x_4) = 2x_1 \cdot x_2 + x_3 \cdot x_4 \quad (7)$$

Izrazi za geometrijske karakteristike poprečnog preseka zavarenog nosača strele konzole koje su neophodne u daljoj analizi date su preko sledećih relacija:

$$I_{x,K} = I_x = \frac{1}{12} \cdot s \cdot h^3 + \frac{1}{6} \cdot b \cdot t^3 + \frac{1}{2} \cdot b \cdot t \cdot (h+t)^2 \quad (8)$$

$$I_{y,K} = I_y = \frac{1}{12} \cdot (h \cdot s^3 + 2 \cdot t \cdot b^3) \quad (9)$$

$$H = h + 2 \cdot t \quad (10)$$

$$W_x = 2 \cdot \frac{I_x}{H} \quad (11)$$

$$W_y = 2 \cdot \frac{I_y}{b} \quad (12)$$

gde su:

I_x, I_y - glavni momenti inercije poprečnog preseka zavarenog nosača strele konzole u x i y pravcu, respektivno,

H - visina zavarenog nosača,

W_x, W_y - otporni momenti inercije poprečnog preseka zavarenog nosača strele konzole u x i y pravcu, respektivno.

4.2. Funkcije ograničenja

U ovom istraživanju tretirane su četiri funkcije ograničenja, i to: čvrstoća zavarenog profila u njegovom naopterećenijem delu, napon u šavu zavarenog nosača, kao i napon u šavu za vezu zavarenog nosača sa okretnim delom i maksimalni ugib kraja konzole.

4.2.1. Kriterijum čvrstoće

Provera čvrstoće se vrši u najopterećenijem delu konstrukcije konzole, na mestu veze I-profila sa okretnim delom. Uticaj donjeg ojačanja je zanemaren, što je na strani sigurnosti.

Maksimalni napon $\sigma_{\max}$ mora biti manji od dopuštenog σ_{dop} , odnosno:

$$\sigma_{\max} = \sqrt{\sigma_z^2 + 3 \cdot \tau_s^2} \leq \sigma_{dop} \quad (13)$$

$$\sigma_{dop} = \frac{R_e}{\nu_1} \quad (14)$$

Sve neophodne relacije određuju se na sledeći način:

$$\sigma_z = \sigma_{zv} + \sigma_{zh} = \frac{M_v}{W_x} + \frac{M_h}{W_y} \quad (15)$$

$$\tau_s = \frac{F}{A} \quad (16)$$

$$F = \gamma \cdot (\psi \cdot Q + m_t) \cdot g \quad (17)$$

$$F_{st} = (Q + m_t) \cdot g \quad (18)$$

$$F_h = k_a \cdot F_{st} \quad (19)$$

$$M_v = F \cdot L_K + \gamma \cdot \frac{q_K \cdot L_K^2}{2} \quad (20)$$

$$L_K = L - a \quad (21)$$

$$M_h = \gamma \cdot \left(F_h \cdot L_K + k_a \cdot \frac{q_K \cdot L_K^2}{2} \right) \quad (22)$$

$$q_K = 1.1 \cdot \rho \cdot g \cdot A \quad (23)$$

gde su:

σ_{k1} - maksimalni napon savijanja,

τ_s - napon smicanja,

γ - koeficijent koji zavisi od pogonske klase dizalice, prema [22],

ψ - dinamički koeficijent, prema [22],

F - dejstvujuća sila od opterećenja,

F_{st} - sila od statičkog opterećenja,

F_h - sila u horizontalnom pravcu,

k_a - dinamički koeficijent opterećenja dizalice u horizontalnoj ravni, prema [22],

M_v, M_h - maksimalni momenti savijanja u vertikalnoj i horizontalnoj ravni,

q_K - specifična težina zavarenog nosača (uvećana 10 %).

Funkcija ograničenja za ovaj kriterijum ima sledeći oblik:

$$g_1 = \sigma_{\max} - \sigma_{dop} \leq 0 \quad (24)$$

4.2.2. Kriterijum napona u šavu nosača

U ovom kriterijumu vrši se provera podužnih ugaonih šavova koji spajaju delove I-nosača (pojasne limove sa vertikalnim limom), [23].

Dokaz napona u ovim šavovima se vrši prema izrazu:

$$\sigma_s = \frac{F \cdot S_{x,s}}{2 \cdot I_x \cdot a_s} \leq \sigma_{s,dop} \quad (25)$$

$$\sigma_{s,dop} = 0.75 \cdot \frac{R_e}{\nu_1} \quad (26)$$

$$S_{x,s} = \frac{b \cdot t}{2} \cdot (h + t) \quad (27)$$

gde su:

σ_s - napon u šavu,

$S_{x,s}$ - statički moment inercije merodavan za proračun šava,

$\sigma_{s,dop}$ - dopušteni napon u šavu.

Funkcija ograničenja u ovom slučaju ima sledeći oblik:

$$g_2 = \sigma_s - \sigma_{s,dop} \leq 0 \quad (28)$$

4.2.3. Kriterijum napona šava za vezu nosača sa okretnim delom

U ovom kriterijumu vrši se provera ugaonih šavova koji spajaju zavareni I-nosač sa okretnim delom (Slika 1 – Detalj “A”). izgled ovog spoja prikazan je na Slici 3a.

Površina šava merodavna za proračun prikazana je na Slici 3b.

Slika 3: Poprečni presek I-profila strele
(a - detalj veze zavarenog nosača sa okretnim delom, b - površina šava merodavna za proračun)

Geometrijski karakteristike konture šava neophodne za analizu određuju se na osnovu sledećih relacija:

$$I_{x,\bar{s}} = \frac{1}{6} \cdot a_{\bar{s}} \cdot h_1^3 + \frac{1}{6} \cdot b \cdot a_{\bar{s}}^3 + \frac{1}{3} \cdot b_1 \cdot a_{\bar{s}}^3 + \frac{1}{2} \cdot b \cdot a_{\bar{s}} \cdot (h + a_{\bar{s}})^2 + b_1 \cdot a_{\bar{s}} \cdot (h - a_{\bar{s}})^2 \quad (29)$$

$$I_{y,\bar{s}} = \frac{1}{6} \cdot a_{\bar{s}} \cdot b^3 + \frac{1}{3} \cdot a_{\bar{s}} \cdot b_1^3 + b_1 \cdot a_{\bar{s}} \cdot (b - b_1)^2 + \frac{1}{6} \cdot h_1 \cdot a_{\bar{s}}^3 + \frac{1}{2} \cdot h_1 \cdot a_{\bar{s}} \cdot (s + a_{\bar{s}})^2 \quad (30)$$

$$W_{x1,\bar{s}} = \frac{I_{x,\bar{s}}}{y_{1,\bar{s}}} \quad (31)$$

$$y_{1,\bar{s}} = \frac{H}{2} + a_{\bar{s}} \quad (32)$$

$$W_{y1,\bar{s}} = \frac{I_{x,\bar{s}}}{x_{1,\bar{s}}} \quad (33)$$

$$x_{1,\bar{s}} = \frac{b}{2} \quad (34)$$

$$W_{x2,\bar{s}} = \frac{I_{x,\bar{s}}}{y_{2,\bar{s}}} \quad (35)$$

$$y_{2,\bar{s}} = \frac{h_1}{2} \quad (36)$$

$$W_{y2,\bar{s}} = \frac{I_{x,\bar{s}}}{x_{2,\bar{s}}} \quad (37)$$

$$x_{2,\bar{s}} = \frac{s}{2} + a_{\bar{s}} \quad (38)$$

$$A_{x,\bar{s}} = 2 \cdot b \cdot a_{\bar{s}} + 4 \cdot b_1 \cdot a_{\bar{s}} \quad (39)$$

$$A_{y,\bar{s}} = 2 \cdot h_1 \cdot a_{\bar{s}} \quad (40)$$

$$b_1 = \frac{1}{2} \cdot (b - 4 \cdot a_{\bar{s}} - s) \quad (41)$$

$$h_1 = h - 4 \cdot a_{\bar{s}} \quad (42)$$

gde su:

$I_{x,\bar{s}}, I_{y,\bar{s}}$ - glavni momenti inercije konture šava u x i y pravcu, respektivno,

$W_{x1,\bar{s}}, W_{y1,\bar{s}}$ - otporni momenti inercije konture šava u x i y pravcu za tačku 1, respektivno,

$W_{x2,\bar{s}}, W_{y2,\bar{s}}$ - otporni momenti inercije konture šava u x i y pravcu za tačku 2, respektivno,

$A_{x,\bar{s}}, A_{y,\bar{s}}$ - površine konture šava merodavne za proračun u x i y , respektivno.

Kontrola napona izvršice se u karaktristični tačkama 1 i 2 konture šava (Slika 3b), pri čemu mora biti ispunjeno, [23]:

$$\sigma_{1,\bar{s}} = \sqrt{\sigma_{z1,\bar{s}}^2 + 3 \cdot V_1^2} \leq \sigma_{\bar{s},dop} \quad (43)$$

$$\sigma_{2,\bar{s}} = \sqrt{\sigma_{z2,\bar{s}}^2 + 3 \cdot V_2^2} \leq \sigma_{\bar{s},dop} \quad (44)$$

Sve neophodne relacije sadržane u prethodnim izrazima određuju se na sledeći način:

$$\sigma_{z1,\bar{s}} = \frac{M_v}{W_{x1,\bar{s}}} + \frac{M_h}{W_{y1,\bar{s}}} \quad (45)$$

$$V_1 = \frac{F_h}{A_{x,\bar{s}}} \quad (46)$$

$$\sigma_{z2,\bar{s}} = \frac{M_v}{W_{x2,\bar{s}}} + \frac{M_h}{W_{y2,\bar{s}}} \quad (47)$$

$$V_2 = \frac{F}{A_{y,\bar{s}}} \quad (48)$$

gde su:

$\sigma_{z1,\bar{s}}, \sigma_{z2,\bar{s}}$ - naponi savijanja šava za tačke 1 i 2, respektivno,

V_1, V_2 - naponi smicanja šava za tačke 1 i 2, respektivno.

Funkcije ograničenja ima sledeći oblik:

$$g_{3,1} = \sigma_{1,\bar{s}} - \sigma_{\bar{s},dop} \leq 0 \quad (49)$$

$$g_{3,2} = \sigma_{2,\bar{s}} - \sigma_{\bar{s},dop} \leq 0 \quad (50)$$

4.2.4. Kriterijum ugiba nosača

Ugib vrha strele konzole f_u (krajnji položaj kolica dizalice) koji mora biti manji od dozvoljenog ugib f_d , određuje se po sledećem izrazu (51), i sastoji se od tri komponente:

$$f_u = f_s + f_{K1} + f_{K2} \leq f_d \quad (51)$$

$$f_d = (H_s + L) \cdot K_f \quad (52)$$

Ugib vrha strele stubne konzole dizalice se računa kao superpozicija ugiba usled uticaja stuba, ugiba vrha strele konzole od dejstva statičke sile (koncentrisano opterećenje na kraju konzole) i ugiba od sopstvene težine (kontinualno opterećenje duž cele konzole).

Kao što se vidi na Slici 1, konzolni deo se posmatra tačno do mesta na kome se stvarno nalazi, pomereno za veličinu a od ose stuba, tako da se ugib ovog dela neće posmatrati, jer se može smatrati dovoljno krutim da se pomera zajedno sa stubom, pre svega zbog same veze koju preko uški i osovine čini sa stubom, kao i zbog veličine samog rastojanja a .

Komponente ugiba se određuju na osnovu sledećih relacija:

$$f_s = tg \left(\frac{M_{v,st} \cdot (H_s - H_1 / 2)}{E \cdot I_{x,s}} \right) \cdot L_K \quad (53)$$

$$M_{v,st} = F_{st} \cdot L_K + \frac{q_K \cdot L_K^2}{2} \quad (54)$$

$$f_{K1} = \frac{F_{st} \cdot L_K^3}{3 \cdot E \cdot I_{x,K}} \quad (55)$$

$$f_{K2} = \frac{q_K \cdot L_K^4}{8 \cdot E \cdot I_{x,K}} \quad (56)$$

gde su:

f_s - ugib usled uticaja konstrukcije stuba,

$M_{v,st}$ - moment savijanja usled statičkog dejstva,

f_{K1} - ugib usled dejstva težina tereta i vitla,

f_{K2} - ugib od sopstvene težine zavarenog I-nosača,

Funkcija ograničenja ima sledeći oblik:

$$g_4 = f_u - f_d \leq 0 \quad (57)$$

5. NUMERIČKI PRIKAZ REZULTATA OPTIMIZACIJE

Postupak optimizacija izvršen je upotrebom sledećih algoritama optimizacije: u MATLAB softverskom paketu korišćenjem pomenutih algoritama ubrzanog roja čestica (APSO), svitca (FA) i kukavičje

pretrage (CSA). Takođe, primenjeni su i GRG2 algoritma i EA algoritma, korišćenjem alata Solver u Ms EXCEL softverskom paketu.

Parametri optimizacije su visina rebra zavarenog nosača h , debljine s , širina pojasnog lima zavarenog nosača b , debljine t , kao i debljina šava zavarenog nosača a_s (Slika 2 i Slika 3).

Pored pomenutih funkcija ograničenja (24), (28), (49), (50) i (57) postoje i dodatna konstruktivna i tehnološka ograničenja koja moraju biti ispunjena.

Što se tiče debljine lima za rebro zavarenog nosača, minimalna debljina iznosi $s_{min}=5 \text{ mm}$, dok je minimalna debljina za lim pojasa zavarenog nosača $t_{min}=6 \text{ mm}$.

Takođe, usvojeno je u ovoj analizi da je minimalna visina rebra $h_{min}=200 \text{ mm}$, dok je minimalna širina pojasa zavarenog nosača $b_{min}=100 \text{ mm}$.

Za debljinu šava se koristi uslov, prema [21]:

$$a_s \leq 0.7 \cdot \min(t, s) \quad (58)$$

Tabela 1: Karakteristike stubnih konzolnih dizalica

Br.	L (m)	a (mm)	H ₁ (m)	I _{x,s} (cm ⁴)	Profil	b (cm)	t (mm)	h (cm)	s (mm)	A _p (cm ²)
1	3	280	0.72	3560	IPE-220	11	9.2	20.16	5.9	33.4
2	5	305	0.90	19870	IPE-270	13.5	10.2	24.96	6.6	45.9

Pored geometrijskih karakteristika stubnih konzolnih dizalica, u prethodnoj tabeli su prikazane i geometrijske karakteristike standardnih IPE profila.

U narednim tabelama prikazani su rezultati optimizacije (optimalni geometrijski parametri poprečnog preseka i optimalna površina poprečnog preseka) prema gorenavedenim algoritmima (Tabela 2 ÷ Tabela 6).

Tabela 2: Rezultati optimizacije primenom APSO algoritma

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.36	6	24.81	5.02	3.51	24.88
2	13.82	6	32.59	5.08	3.56	33.14

Tabela 3: Rezultati optimizacije primenom FA algoritma

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	11.79	6	21.07	5.19	3.63	25.09
2	15.87	6	28.51	5	3.5	33.30

Tabela 4: Rezultati optimizacije primenom CSA algoritma

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.66	6	24.12	5	3.5	24.86
2	14.39	6	31.62	5	3.5	33.08

Tabela 7: Rezultati optimizacije primenom APSO algoritma i ostvarena ušteda

Br.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Ušteda (%)
1	10.4	6	24.9	6	4	27.42	17.90
2	13.9	6	32.6	6	4	36.24	21.05

Tabela 8: Rezultati optimizacije primenom FA algoritma i ostvarena ušteda

Br.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Ušteda (%)
1	11.8	6	21.1	6	4	26.82	19.70
2	15.9	6	28.6	5	4	33.38	27.28

Tabela 9: Rezultati optimizacije primenom CSA algoritma i ostvarena ušteda

Br.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Ušteda (%)
1	10.7	6	24.2	5	4	24.94	25.33
2	14.4	6	31.7	5	4	33.13	27.82

Takođe, minimalna vrednost debljine šava iznosi $a_s=3 \text{ mm}$, [23].

Predstavljeni model optimizacije biće sproveden na primerima dve stubne konzolne dizalice, koje se nalaze u eksploataciji. Za konstrukcije strele obe dizalice korišćeni su standardni IPE profili od običnog konstruktivnog čelika S235.

Obe dizalice koristi električno lančano vitlo sa kojim se upravlja sa poda. Okretanje strele se vrši ručno, preko lanca.

Koeficijenti koje se uzimaju za optimizaciju imaju sledeće vrednosti:

$$\gamma=1.05, \psi=1.15, k_a=0.05, K_f=1/250$$

Obe dizalice su druge pogonske klase, nosivosti $Q=500 \text{ kg}$, težine vitla $m_t=40 \text{ kg}$ i visine stuba $H_s=4 \text{ m}$. Ostali podaci za dizalice su prikazani u sledećoj tabeli (Tabela 1):

Tabela 5: Rezultati optimizacije primenom GRG2 algoritma

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.66	6	23.37	5	3.5	24.48
2	14.48	6	30.63	5	3.5	32.69

Tabela 6: Rezultati optimizacije primenom EA algoritma

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.54	6	23.44	5.06	3.54	24.50
2	14.23	6	30.94	5.06	3.54	32.71

Kao što se može videti na osnovu dobijenih rezultata, u nekim slučajevima nisu dobije cele vrednosti za debljine limova, tako da su ove vrednosti zaokružene na prvu veću celu vrednost. Takođe, i ostale vrednosti su zaokružene. Na ovaj način se stiče realna slika o dobijenim rezultatima.

U narednim tabelama prikazane su nove zaokružene vrednosti za optimalne geometrijske parametre poprečnog preseka, optimalne površine poprečnog preseka i ostvarena ušteda u odnosu na standardne površine profila (Tabela 7 ÷ Tabela 11):

Tabela 10: Rezultati optimizacije primenom GRG2 algoritma i ostvarena ušteda

Br.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Ušteda (%)
1	10.7	6	23.4	5	4	24.54	26.53
2	14.5	6	30.7	5	4	32.75	28.65

Tabela 11: Rezultati optimizacije primenom EA algoritma i ostvarena ušteda

Br.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Ušteda (%)
1	10.6	6	23.5	6	4	26.82	19.70
2	14.3	6	31.0	6	4	35.76	22.09

Iz prethodnih tabela (Tabela 7 ÷ Tabela 11) se vidi da su sada nove optimalne površine veće od onih prikazanih u Tabela 2 ÷ Tabela 7. Kod određenih algoritama optimizacije je moguće uvesti ugraničenje da varijabla uzima celu vrednost. U ovom slučaju je to moguće ostvariti kod primene alata Solver u Ms EXCEL softverskom paketu, kod GRG2 i EA algoritma optimizacije, korišćenjem opcije integer, ali je određeno na dati način kako bi rezultati mogli da se porede sa onima dobijenim algoritmima u MATLAB softverskom paketu.

6. ZAKLJUČAK

U radu su određene optimalne geometrijske vrednosti za zavareni nosač I-poprečnog preseka strele stubne konzolne dizalice, na primerima dve stubne konzolne dizalice nosivosti 500 kg. Kao objekat optimizacije posmatrana je površina poprečnog preseka konzolnog nosača, pri čemu su sve funkcije ograničenja zadovoljene. Pored glavnih funkcija ograničenja, čvrstoće zavarenog I-profil, napon u šavu zavarenog nosača, napon u konturi šava u karakterističnim tačkama za vezu zavarenog nosača, kao i dopuštenog ugib kraja konzole, korišćeni su i dodatni konstruktivni i tehnološki kriterijumi. Optimizacija je sprovedena korišćenjem MATLAB softverskog paketa, primenom algoritma ubrzanog roja čestica (APSO), algoritma svitca (FA) i algoritma kukavičje pretrage (CSA), kao i korišćenjem Ms EXCEL softverskog paketa, primenom GRG2 algoritma i EA algoritma optimizacije posredstvom alata Solver.

Izbor odgovarajućih metoda optimizacije pokazuje njihovu opravdanost, jer je ostvarena ušteda od 28.65 %, na primeru prve dizalice i 26.53 %, na primeru druge dizalice.

Postavljeni zadatak optimizacije, smanjenje mase zavarenog nosača I-poprečnog preseka, je uspešno izvršen, što se vidi na osnovu rezultata u prethodnim tabelama (Tabela 2 ÷ Tabela 11) i pokazuje ispravnost korišćenja predstavljenog modela za analizu i optimizaciju.

Na osnovu rezultata iz Tabela 2 ÷ Tabela 6, vidi se da je najbolje rezultate dao GRG2 algoritam, a najslabije algoritam svitca (FA), i na jednom i na drugom korišćenom primeru. Što se tiče prirodom inspirisan algoritama, najbolje rezultate je da algoritam kukavičje pretrage (CSA). Takođe, na osnovu ovih tabela se vidi da je u svim slučajevima kod debljine pojasnog lima t , dobijena minimalno postavljena granična vrednost ove debljine (6 mm).

Što se tiče korišćenih metoda optimizacije u ovom istraživanju, najbolje rezultate daje GRG2, kod upotrebe alata Solver u Ms EXCEL softverskom paketu, odnosno algoritam kukavičje pretrage (CSA), korišćenjem MATLAB softverskog paketa (Tabela 2 ÷ Tabela 6). Ovo

se takođe vidi na osnovu tabela gde su zaokružene vrednosti dobijenih rezultata (Tabela 7 ÷ Tabela 11).

Glavni zaključak na osnovu sprovedene analize i optimizacije je da zavareni nosač I-poprečnog preseka daje značajne uštede u materijalu u odnosu na standardne valjane I-profile koji se najčešće primenjuju.

Za buduća istraživanja u ovoj oblasti, neophodno je uključiti i druge funkcije ograničenja, koje su od značaja za analizu. Pored ugiba i naponskih stanja u materijalu i šavovima, može se analizirati i elastična stabilnost nosača. Takođe, mogu se analizirati i drugi oblici poprečnih preseka nosača koji se koriste za ove tipove struktura, kao i drugi potencijalni oblici poprečnih preseka koji mogu doći u obzir, kao i tipovi čelika.

Kao moguća funkcija cilja, u daljijim istraživanjima se može uzeti u obzir i površina konture šava, pored površine poprečnog preseka, tako da bi se u tom slučaju primenio neki od multi-objektivnih postupaka optimizacije. Cena koštanja izrade celog nosača može biti isto predlog kao funkcija cilja u procesu optimizacije.

Rezultati dobijeni na ovaj način se mogu verifikovati i porediti sa rezultatima dobijenim na osnovu MKE analize, u nekom od softverskih paketa, kako bi se pokazala opravdanost primene datih metoda. Takođe, na ovaj način se mogu izvesti određeni zaključci i dati smernice kod projektovanja i optimizacije ovakvog tipa konstrukcija. Optimizacija se može izvršiti i hibridnim metodama, kombinacijom nekih od numeričkih postupaka optimizacije sa MKE softverskim paketom.

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LITERATURA

- [1] R.K. Amreeta and V. Singh, "Design and Stress Analysis of Single Girder Jib Crane", International Journal of Engineering Research & Technology (IJERT), Vol. 4, Issue 09, pp. 932-936, (2015)
- [2] S.N. Khetre, P.S. Bankar and A.M. Meshram, "Design and Static Analysis of I-Section Boom for Rotary Jib Crane", International Journal of Engineering Research & Technology (IJERT), Vol. 3, Issue 8, pp. 1071-1074, (2014)
- [3] M. Dhanoosha and V.G. Reddy "Detail Design and Analysis of A Free Standing I Beam Jib Crane", International Research Journal of Engineering and Technology (IRJET), Vol. 3, Issue 12, pp. 193-203, (2016)

- [4] D. Desai and A.N. Patil, "Analysis & Optimization of Jib Crane under Influence of Various Slewing angles & load position by limit state method", International Engineering Research Journal, pp. 1207-1215
- [5] G. Pavlović, M. Gašić, M. Savković, N. Zdravković and G. Marković, "Comparative Analysis of the Model for Determination of Deflection in the Column-Mounted Jib Crane Structure", XXIII INTERNATIONAL SCIENTIFIC CONFERENCE "TRANSPORT 2017", Sofia (Bulgaria), 12–14 October, Vol. 15, Issue 3, article No. 1528, (2017)
- [6] S.S.Raut and B.C. Londhe, "Design and Analysis of Column Structure for Jib Crane", International Journal for Educational Research and Interdisciplinary Studies (IJERIS), Vol. VI, Issue XI, pp. 1-6, (2017)
- [7] S.D. Kurhekar and R. Gulpude, "Design Optimization and Finite Element Analysis of Movable Jib Crane", International Journal of Analytical, Experimental and Finite Element Analysis (IJAEFEA), Vol. 2, Issue 3, pp. 112-114, (2015)
- [8] X. Ma and X. Cheng, "Model Building of Finite Element Analysis Based on ANSYS of Column Jib Crane", Trans Tech Publications, Vol. 462, pp. 427-433, (2012)
- [9] T. Patil and N.L. Shelke, "Structural Analysis of a Cantilever Beam with Tapered Web Section through FEA", Journal of Steel Structures & Construction, Vol. 2, Issue 2, 1000119, (2016)
- [10] K.S. Bollimpelli and V.R. Kumar, "Design and Analysis of Column Mounted JIB Crane", International Journal of Research in Aeronautical and Mechanical Engineering (IJRAME), Vol. 3, Issue 1, pp. 32-52, (2015)
- [11] V.S. Chavan and S.S. Waydande, "Design optimization and Modal Analysis of Cantilever I-section Beam For 0.5 ton capacity of Floor Mounted Jib Crane", International Journal of Mechanical and Industrial Technology, Vol. 5, Issue 1, pp. 85-99, (2017)
- [12] T. Haniszewski and D. Gąska, "Numerical Modelling of I-Beam Jib Crane with Local Stresses in Wheel Supporting Flanges - Influence of Hoisting Speed", Naše more, Vol. 64, Issue 1, pp. 7-13, (2017)
- [13] P.S. Padwal and A.R. Patil, "Review & Analysis of Cantilever Beam Crane for Weight Reduction by Using Topography Method", International Journal of Advance Research and Innovative Ideas in Education (IJARIE), Vol. 3, Issue 4, pp. 1797-1805, (2017)
- [14] C.C. Dandavatimath and H.D. Sarode, "Finite Element Analysis and Optimization of Jib Crane Boom", International Journal of Innovative Research in Science, Engineering and Technology (IJIRSET), Vol. 6, Issue 7, pp. 14287-14294, (2017)
- [15] K. Gandhare and V. Thute, "Design Optimization of Jib Crane Boom Using Evolutionary Algorithm", International Journal of Scientific Engineering and Research (IJSER), Vol. 3, Issue 4, Paper ID: IJSER1553, (2015)
- [16] S. Wu and T. Su, "Optimization Design of Cantilever Beam for Cantilever Crane Based on Improved GA", TELKOMNIKA Indonesian Journal of Electrical Engineering, Vol. 12, No. 4, pp. 2652-2657, (2014)
- [17] G. Pavlović, A. Stepanović, J. Vidaković, M. Savković and N. Zdravković, "Design Optimization of the Box Section of the Single-Girder Bridge Crane by Generalized Reduced Gradient Algorithm", IMK-14, Research & Development in Heavy Machinery, Vol. 21, No. 3, pp. EN79-EN86, (2015)
- [18] G. Pavlović, M. Gašić, M. Savković and N. Zdravković, "Comparative Analysis of Local and Lateral Stability of Plates as the Constraint Functions Within Optimization of Main Girder Box Section of the Bridge Crane", IMK-14 – Research & Development, Vol. 18, No. 1, pp. EN11-EN18, (2012)
- [19] <https://www.mathworks.com/matlabcentral/fileexchange/29725-accelerated-particle-swarm-optimization>
- [20] https://www.mathworks.com/matlabcentral/fileexchange/29693-firefly-algorithm?s_tid=srchtitle
- [21] <https://www.mathworks.com/matlabcentral/fileexchange/29809-cuckoo-search-cs-algorithm>
- [22] D. Ostrić and S. Tošić, "Dizalice", Institut za mehanizaciju Mašinskog fakulteta Univerziteta u Beogradu, Beograd, (Srbija), (2005)
- [23] Z. Petković and D. Ostrić, "Metalne konstrukcije u mašogradnji I", Institut za mehanizaciju Mašinskog Fakulteta Univerziteta u Beogradu (Srbija), (1996)

Optimal Design of Welded I-beam of Slewing Pillar Jib Crane

Goran Pavlović^{1*}, Mile Savković², Goran Marković², Nebojša Zdravković², Jelena Stanojković¹

¹ Research and development center Alfatec d.o.o., Niš (Srbija)

² Faculty of Mechanical and Civil Engineering in Kraljevo, University of Kragujevac, Kraljevo (Serbia)

In this paper, the process of analyzing and optimizing of jib boom is presented. The reduction in the weight of the welded I-beam, or the cross-sectional area of the beam, was set as the main objective of this study, with the essence of replacing the standard rolled profile with welded ones. Apart from the criteria of allowed deflection and stress, stresses in a weld, both in the carrier itself and in carrier connection with a swivel part of structure. Also, certain constructive and technological recommendations were used, which representing additional constraint function in the optimization process. Optimization was performed using by MATLAB software package for Adaptive Particle Swarm Optimization algorithm (APSO), the Firefly Algorithm (FA) and the Cuckoo Search Algorithm (CSA), as well as the Ms. EXCEL software package, i.e. Solver tool, where optimization algorithms for constrained nonlinear problems are used (the generalized reduced gradient method and the evolution optimization algorithm). The obtained results were verified on real examples of pillar jib cranes, and they are showing the justification of procedure application.

Keywords: Pillar jib crane, Matlab, Ms Excel, Optimization, Welded carrier

1. INTRODUCTION

Today's industry requires versatile, efficient and cost-effective equipment, while at the same time it should provide more flexibility with significant savings through increased productivity. Pillar jib cranes can significantly help to improve the efficiency of manipulation with materials and work and production flows. More significant consideration should be given to an operational environment which requires more frequent repetition of lifting and transferring of loads within the fixed arc of rotation.

The need for continual improvement in material manipulation technologies remains a typical feature of many modern engineering problems. The right equipment selection and dimensioning entire system segments are most significant. All the necessary conditions must be met in order to achieve equipment stability and safety and the costs which should also be taken into account. Therefore, comprehensive analysis and optimization of design parameters of jib cranes are necessary. A reduction in the number of services should be ensured, as well as the undisturbed functioning of the equipment without interruptions in operation.

The cantilever beam of jib crane represents the most responsible part of the structure, and it is the subject of analysis and research in this paper. The construction of the column is not a topic of analysis and optimization since it is a segment made of a standard welded tube. The optimization of this geometry is not significant for the applied optimization procedure in this research, due to the economic aspect.

Optimization represents a process where the most superior values of the parameters (variables) are obtained based on the given constraint functions, for the observed objective function. The most common are optimization procedures aimed at minimizing mass and cost. Latterly, the most common have implemented the methods of multi-objective optimizations, where several objective functions

exist in the optimization process. For the above reasons and due to the importance of these specific types of constructions, a large number of surveys and publications dealing with the problems of structures analysis and optimization of these cranes, and especially the jib boom structure as its most important and most responsible segment.

The most primary criterion in the process of analysis and optimization represent the deflection at end of the span, as well as the stress conditions of the structure. The analysis of the deflection and stress conditions is done in most cases using the FEM, whereby the results are compared with analytical ([1], [2], [3], [4], [6], [10] and [11]). In the paper [1], an analysis of the deflection and stress conditions of the jib boom was performed using the ANSYS software package, where the 3D model was implemented in the CATIA software package. In addition to the typical standard I-beam, which is most commonly used in these constructions, a standard box profile is also observed. Similar to the previous one, in the work [2], using the COSMOS software package, the analysis of jib boom in a form of I-beam was performed, while the 3D model was generated in the SOLIDWORKS software package.

In published articles [3] and [4], for the key difference to the previously mentioned works, the overall structures of the jib cranes (column and cantilever) were analyzed. In the paper [3], various types of steel were analyzed for the case of one column jib crane using the ANSYS software package, while the 3D model was made in the Pro/E software package. Specific recommendations for the choice of steel type are given, considering the concrete conditions in crane operation. In the paper [4], the authors carried out a comprehensive analysis of the jib crane structure, for various positions of the load and the boom angle, where the allowed deflection is set as the primary criterion, whereby the results were verified experimentally.

*Corresponding author: Bulevar Nikole Tesle 63/5, 18000 Niš, Srbija, email:goran.pavlovic@alfatec.rs

FME analysis was performed using the and ANSYS software packages. The deflection at end of the span was analyzed in detail in [5], for different calculation models, where the results were verified using the CATIA and SAP2000 software packages. A detailed analysis of the column structure was performed by NX 10.0 [6]. Using the Inventor software package, in [7], the structure optimization of a wall mounted jib crane, is performed in order to reduce the mass of the existing structure.

The importance of appropriate modeling and structural analysis using the FEM was demonstrated in [8], where the procedure for modeling of the jib crane assembly is performed in the ANSYS software package.

In previously mentioned researches, the standard I-beam which is commonly used was observed. In the paper [9], by application of the ANSYS software package is analyzed the variable (conical) I-beam, for different conical, angular values and taking into account the deflections, stresses, and the grinder elastic stability.

In addition to deflections and stresses, some authors assert the importance of modular analysis of crane structure ([10], [11] and [12]). In the paper [10], with the application of the ANSYS software package is analyzed a jib crane structure, showing the frequency dependence on stresses and deflections in certain directions. In the paper [11], the optimal type of I-beam is determined by looking at several selected types of standard I profile, and results are verified experimentally. In the study [12], the authors present a modeling the dynamics of load lifting and its impact on the I-beam of the jib crane boom, where the simulation was performed implementing the MATLAB-Simulink software.

The topological analyzes and optimizations are increasingly being applied, and their significance and application are shown in the review paper [13], for the I-profile and in the double I-profiles. In addition, in the paper [14], various rib profiles and its parameters were varied on the I-beam, and influence on the deflection and stress states were monitored. In contrast to the above-mentioned publications, where FEM was applied, in the works [15] and [16] optimization of the boom cross-sectional area is performed using certain numerical optimization algorithms. In the paper [15], the optimization of the box-shaped cross-sectional area is performed using the evolutionary optimization algorithm (EA) in the EXCEL software package. The significant savings were achieved in comparison with the existing solution. Using the improved genetic optimization algorithm (IGA), within [16], the geometric parameters of the jib boom structure were optimized and where the savings of about 20% was achieved.

In addition, the EXCEL software package has been successfully used in the crane structures optimization of the overhead crane with one girder, as shown in [17]. In the paper [18], the structure optimization of the two-girder overhead crane was carried out, using the analytical optimization method. The explicit value of the expression for the optimal geometric parameters and their relations were obtained.

Ultimately, the objective of this research is the analysis and multi-criteria optimization of the welded I-beam geometric parameters.

Analysis and optimization will be carried out on examples of jib cranes that are in exploitation.

2. APPLIED OPTIMIZATION ALGORITHMS

Various numerical methods (algorithms) of optimization are used in this research, using by MATLAB and EXCEL software packages. The metaheuristic optimization algorithms, Adaptive Particle Swarm Optimization algorithm (APSO), Firefly Algorithm (FA), and Cuckoo Search Algorithm (CSA) in the MATLAB software package are applied. Equally, the Generalized Reduced Gradient Algorithm (GRG2) and the Evolution Algorithm (EA) in the EXCEL software package are used.

The Adaptive Particle Swarm Optimization algorithm, Firefly Algorithm Cuckoo Search Algorithm optimization algorithms. The Adaptive Particle Swarm Optimization algorithm and the Firefly algorithm were developed in 2008 by X. S. Yang. In 2009, the identical author together with S. Deb presented the Cuckoo Search Algorithm. All these three algorithms were applied in this study in their source code ([19], [20] and [21]). EXCEL Solver implements the generalized reduced gradient method (GRG2 algorithm) to optimize non-linear problems and was developed by L. Lasdon and A. Varen, 2011. Equally, Solver additionally implements an evolutionary algorithm (EA) to solve nonlinear optimization problems.

3. MATHEMATICAL FORMULATION OF OPTIMIZATION PROBLEM

The optimization process is aimed at determining the optimum geometric parameters of the welded I-beam, which will lead to a reduction in its mass and the cross-sectional area. The optimization problem is defined as follows:

$$\text{minimizing of an objective function} \\ f(X) \quad (1)$$

in relation to the constraint functions

$$g_i(X) \leq 0 \quad (2)$$

Also, the following conditions must be fulfilled:

$$l_i \leq X_i \leq u_i \quad (3)$$

where are:

$f(X)$ - objective function,

$g_i(X) \leq 0, i = 1, \dots, m$ - constraint functions,

l_j, u_j - lower and upper limit of a variable,

$i = 1, \dots, m$ - number of constraint functions,

$j = 1, \dots, n$ - number of projected variables,

X - project vector consisting of n projected variables.

Project variables are the values which do have to determine during the optimization process (each project variable is defined by its lower and upper limit).

To define the objective function and constraint functions, firstly the engineering problem that is the basis of this research will be presented. In this paper, the main subjects of research are related to the analysis and optimization of the I-beam structure (Figure 1).

The main idea is to thoroughly analyze and optimize the welded I-beam or to demonstrate the justification of making such a carrier that would replace the standard INP

or IPE profiles which are most commonly used in these types of structures to reduce the carrier weight.

In Figure 1, one type of pillar jib crane and the basic input geometric parameters are shown. The structure consists of two basic parts, column structure, with height H_s and diameter D_k , and boom structure, with length L_k , which rotates around the axis of the column, over the axle with the length H_1 , at distance a from the axis of the column. The hoist trolley is taken in the analysis at end of the span, or at a distance L from the axis of the column (the most unfavorable position).

Figure 1: Pillar jib crane

The input values that are necessary for this analysis are shown via the vector of input parameters:

$$\vec{x} = (Q, m_t, L, a, H_s, H_1, I_{x,s}, R_e, K_f \dots) \quad (4)$$

where are:

Q - load capacity,

m_t - hoist trolley weight,

L - the arc length of crane rotation,

$I_{x,s}$ - the main moment of area of the column,

R_e - material yield strength,

K_f - the coefficient that depends on the mode of crane operation and the drive class, according to [22].

The vector of the optimization parameter (projected variables) is:

$$X = (x_1 \ x_2 \ x_3 \ x_4 \ x_5)^T = (b \ t \ h \ s \ a_s)^T \quad (5)$$

The following text will show detailed optimization parameters (variables), objective function and constraint functions.

3. OBJECTIVE FUNCTION AND CONSTRAINT FUNCTIONS

4.1 Objective function

The objective function represents the I-beam cross-sectional area for an observed welded carrier of jib boom (Figure 2).

Figure 2: Welded I- beam of the jib boom

The cross-sectional area of the welded carrier, or the objective function is given by:

$$A = 2 \cdot b \cdot t + h \cdot s \quad (6)$$

where are:

b - profile width of the welded carrier,

t - flange sheet thickness of the welded carrier,

h - web height of welded carrier,

s - web thickness of welded carrier.

The above-mentioned values typically represent the project variables in the optimization process. The weld thickness does not figure in the objective function, and its influence is indirect on the optimal parameters of the I-beam cross-sectional area, across the geometric characteristics which influencing in the criteria in where the stress in the weld exists. The identical value of weld thickness was adopted for both of these criteria.

The mathematical formulation of the target function is shown as follows:

$$f(X) = A(x_1 \ x_2 \ x_3 \ x_4) = 2x_1 \cdot x_2 + x_3 \cdot x_4 \quad (7)$$

The expressions of cross-sectional area geometric characteristics of jib boom with the welded carrier which are necessary for further analysis are given through the following relations:

$$I_{x,K} = I_x = \frac{1}{12} \cdot s \cdot h^3 + \frac{1}{6} \cdot b \cdot t^3 + \frac{1}{2} \cdot b \cdot t \cdot (h+t)^2 \quad (8)$$

$$I_{y,K} = I_y = \frac{1}{12} \cdot (h \cdot s^3 + 2 \cdot t \cdot b^3) \quad (9)$$

$$H = h + 2 \cdot t \quad (10)$$

$$W_x = 2 \cdot \frac{I_x}{H} \quad (11)$$

$$W_y = 2 \cdot \frac{I_y}{b} \quad (12)$$

where are:

I_x, I_y - the planar moments of inertia of welded jib boom in the x and y directions, respectively

H - a height of the welded carrier,
 W_x, W_y - the resistant moment of welded jib boom about the axis x and y , respectively.

4.2 Constraint functions

In this research, four constraints were treated: the strength of the welded profile in its overloaded part, the stress in, as well as the stress in the weld for carrier connection with the swivel part and the maximum deflection of the cantilever at end of the span.

4.2.1 Strength criterion

The strength test is performed in the most heavily loaded part of the cantilever construction, at the proper place of I-beam connection with the swivel part. The effect of the lower reinforcement is neglected, which is precisely on the security side.

The maximum stress $\sigma_{\max}$ must be less than the allowed σ_{dop} , respectively:

$$\sigma_{\max} = \sqrt{\sigma_z^2 + 3 \cdot \tau_s^2} \leq \sigma_{\text{dop}} \quad (13)$$

$$\sigma_{\text{dop}} = \frac{R_e}{\nu_1} \quad (14)$$

All necessary relations are determined as follows:

$$\sigma_z = \sigma_{zv} + \sigma_{zh} = \frac{M_v}{W_x} + \frac{M_h}{W_y} \quad (15)$$

$$\tau_s = \frac{F}{A} \quad (16)$$

$$F = \gamma \cdot (\psi \cdot Q + m_t) \cdot g \quad (17)$$

$$F_{st} = (Q + m_t) \cdot g \quad (18)$$

$$F_h = k_a \cdot F_{st} \quad (19)$$

$$M_v = F \cdot L_K + \gamma \cdot \frac{q_K \cdot L_K^2}{2} \quad (20)$$

$$L_K = L - a \quad (21)$$

$$M_h = \gamma \cdot \left(F_h \cdot L_K + k_a \cdot \frac{q_K \cdot L_K^2}{2} \right) \quad (22)$$

$$q_K = 1.1 \cdot \rho \cdot g \cdot A \quad (23)$$

where are:

σ_{k1} - the maximum bending stress,

τ_s - the shear stress,

γ - coefficient that depends on the driving class of the crane, [22],

ψ - dynamic coefficient, [22],

F - load,

F_{st} - statical load,

F_h - a horizontal force,

k_a - dynamic load coefficient of the crane in the horizontal plane [22],

M_v, M_h - the maximum bending stress in vertical and horizontal plane,

q_K - the specific weight of the welded carrier (increased for 10 %).

The constraint function for this criterion has the following form:

$$g_1 = \sigma_{\max} - \sigma_{\text{dop}} \leq 0 \quad (24)$$

4.2.1 Stress criterion of beam weld

In this criterion, checking the longitudinal angular weld that connects the parts of I-beam (flange sheets with a vertical sheet) is performed, [23]. The stress comparing is given as follows:

$$\sigma_s = \frac{F \cdot S_x}{2 \cdot I_x \cdot a_s} \leq \sigma_{s,\text{dop}} \quad (25)$$

$$\sigma_{s,\text{dop}} = 0.75 \cdot \frac{R_e}{\nu_1} \quad (26)$$

$$S_{x,s} = \frac{b \cdot t}{2} \cdot (h + t) \quad (27)$$

where are:

σ_s - the stress in the weld,

$S_{x,s}$ - section modulus appropriate for weld calculation,

$\sigma_{s,\text{dop}}$ - the allowed stress in weld.

The constraint function in this case has the following form:

$$g_2 = \sigma_s - \sigma_{s,\text{dop}} \leq 0 \quad (28)$$

4.2.2 Stress criterion of weld for connection between beam and swivel part

In this criterion, checking the angular welds that connect the welded I-beam with the swivel part (Figure 1 - Detail "A"). The design of this compound is shown in Figure 3a. The welds area which is relevant for the calculation is given in Figure 3b.

Figure 3: I-beam cross section
(a – detail of connection between the welded beam and swivel part, b – the area of the weld)

The geometrical characteristics of the weld contour necessary for analysis are determined on the basis of the following relations:

$$I_{x,s} = \frac{1}{6} \cdot a_s \cdot h_1^3 + \frac{1}{6} \cdot b \cdot a_s^3 + \frac{1}{3} \cdot b_1 \cdot a_s^3 + \frac{1}{2} \cdot b \cdot a_s \cdot (h + a_s)^2 + b_1 \cdot a_s \cdot (h - a_s)^2 \quad (29)$$

$$I_{y,s} = \frac{1}{6} \cdot a_s \cdot b^3 + \frac{1}{3} \cdot a_s \cdot b_1^3 + b_1 \cdot a_s \cdot (b - b_1)^2 + \frac{1}{6} \cdot h_1 \cdot a_s^3 + \frac{1}{2} \cdot h_1 \cdot a_s \cdot (s + a_s)^2 \quad (30)$$

$$W_{x1,s} = \frac{I_{x,s}}{y_{1,s}} \quad (31)$$

$$y_{1,s} = \frac{H}{2} + a_s \quad (32)$$

$$W_{y1,s} = \frac{I_{x,s}}{x_{1,s}} \quad (33)$$

$$x_{1,s} = \frac{b}{2} \quad (34)$$

$$W_{x2,s} = \frac{I_{x,s}}{y_{2,s}} \quad (35)$$

$$y_{2,s} = \frac{h_1}{2} \quad (36)$$

$$W_{y2,s} = \frac{I_{x,s}}{x_{2,s}} \quad (37)$$

$$x_{2,s} = \frac{s}{2} + a_s \quad (38)$$

$$A_{x,s} = 2 \cdot b \cdot a_s + 4 \cdot b_1 \cdot a_s \quad (39)$$

$$A_{y,s} = 2 \cdot h_1 \cdot a_s \quad (40)$$

$$b_1 = \frac{1}{2} \cdot (b - 4 \cdot a_s - s) \quad (41)$$

$$h_1 = h - 4 \cdot a_s \quad (42)$$

where are:

$I_{x,s}$, $I_{y,s}$ - the planar moments of inertia of the weld contour in x and y direction,

$W_{x1,s}$, $W_{y1,s}$ - the resistant moment of the weld contour in x and y direction for point 1,

$W_{x2,s}$, $W_{y2,s}$ - the resistant moment of the weld contour in x and y direction for point 2,

$A_{x,s}$, $A_{y,s}$ - the area of the weld contour in x and y direction.

The stress comparing will be performed in points 1 and 2 of the weld contour (Figure 3b), and must be fulfilled, [23]:

$$\sigma_{1,s} = \sqrt{\sigma_{z1,s}^2 + 3 \cdot V_1^2} \leq \sigma_{s,dop} \quad (43)$$

$$\sigma_{2,s} = \sqrt{\sigma_{z2,s}^2 + 3 \cdot V_2^2} \leq \sigma_{s,dop} \quad (44)$$

All the necessary relations contained in the above-mentioned terms are determined as follows:

$$\sigma_{z1,s} = \frac{M_v}{W_{x1,s}} + \frac{M_h}{W_{y1,s}} \quad (45)$$

$$V_1 = \frac{F_h}{A_{x,s}} \quad (46)$$

$$\sigma_{z2,s} = \frac{M_v}{W_{x2,s}} + \frac{M_h}{W_{y2,s}} \quad (47)$$

$$V_2 = \frac{F}{A_{y,s}} \quad (48)$$

where are:

$\sigma_{z1,s}$, $\sigma_{z2,s}$ - the bending stress of the weld in point 1 and 2,

V_1 , V_2 - the shear stress of the weld for points 1 and 2.

The constraint functions are given as follows

$$g_{3,1} = \sigma_{1,s} - \sigma_{s,dop} \leq 0 \quad (49)$$

$$g_{3,2} = \sigma_{2,s} - \sigma_{s,dop} \leq 0 \quad (50)$$

4.2.3 A criterion of beam deflection

The deflection of cantilever top *fu* (at end position of the hoist trolley) which must be less than the allowed one f_d is determined by the following expression (51), and consists of three components:

$$f_u = f_s + f_{K1} + f_{K2} \leq f_d \quad (51)$$

$$f_d = (H_s + L) \cdot K_f \quad (52)$$

The cantilever deflection of jib crane at the top is accurately calculated as the superposition of the deflection due to the impact of the column, the boom deflection at the top causes by static force (concentrated load at end of the span) and deflection due to cantilever weights.

As seen in Figure 1, the cantilever is observed exactly to the point at which it is actually located, shifted by the value a from the axis of the column, so that the deflection of this part is unobserved, because it can be considered sufficiently rigid to move along with the column, primarily because of the connection with the column itself as well as the size of the value a .

The deflection components are determined based on the following relations:

$$f_s = tg \left(\frac{M_{v,st} \cdot (H_s - H_1/2)}{E \cdot I_{x,s}} \right) \cdot L_K \quad (53)$$

$$M_{v,st} = F_{st} \cdot L_K + \frac{q_K \cdot L_K^2}{2} \quad (54)$$

$$f_{K1} = \frac{F_{st} \cdot L_K^3}{3 \cdot E \cdot I_{x,K}} \quad (55)$$

$$f_{K2} = \frac{q_K \cdot L_K^4}{8 \cdot E \cdot I_{x,K}} \quad (56)$$

where are:

f_s - deflection due to column structure influence,

$M_{v,st}$ - the moment of bending due to statical load,

f_{K1} - deflection due to load weight and hoist trolley weight,

f_{K2} - deflection due to welded I-beam weights.

The constraint function is given as follows:

$$g_4 = f_u - f_d \leq 0 \quad (57)$$

5. NUMERICAL REVIEW OF OPTIMIZATION RESULT

The optimization process was performed using the following optimization algorithms: in the MATLAB software package using the Adaptive Particle Swarm

Optimization algorithm, Firefly Algorithm, and Cuckoo Search Algorithm. Also, the GRG2 algorithm and EA algorithm were applied, using the Solver tool in the Ms EXCEL software package. Optimization parameters are the rib height of the welded carrier h , the thickness s , the width of flanges b and thickness t , and the weld thickness a_s (Figure 2 and Figure 3).

In addition to the mentioned constraint functions (24), (28), (49), (50) and (57) there are additional constructive and technological constraints that must be met.

Regarding the thickness of the sheet metal for the welded carrier web, the minimum thickness must be $s_{min}=5$ mm, while the minimum thickness of flanges is $t_{min}=6$ mm.

It was also adopted in this analysis that the minimum web height is $h_{min}=200$ mm, while the minimum width of flanges is $b_{min}=100$ mm.

For the thickness of weld is used requirements, according to [21]:

$$a_s \leq 0.7 \cdot \min(t, s) \quad (58)$$

Also, the minimum value of the weld thickness is $a_s=3$ mm, [23].

The presented optimization model will be implemented on examples of two pillar jib cranes, which are in exploitation. For cantilever structures, both cranes used standard IPE profiles made from conventional S235 structural steel.

Both cranes use an electrically hoist trolley with floor control. Turning the jib arm is done manually, over the chain. The coefficients taken for optimization have the following values:

$$\gamma = 1.05, \psi = 1.15, k_a = 0.05, K_f = 1/250$$

Both cranes are second propulsion classes with a load capacity of $Q = 500$ kg, hoist trolley weight $m_t = 40$ kg and column height $H_s = 4$ m. Other data for cranes are shown in the following table (Table 1):

Table 1: Characteristics of pillar jib crane

No.	L (m)	a (mm)	H ₁ (m)	I _{x,s} (cm ⁴)	Profil	b (cm)	t (mm)	h (cm)	s (mm)	A _p (cm ²)
1	3	280	0.72	3560	IPE-220	11	9.2	20.16	5.9	33.4
2	5	305	0.90	19870	IPE-270	13.5	10.2	24.96	6.6	45.9

In addition to the geometrical characteristics of pillar jib cranes, the previous table also shows the geometric characteristics of standard IPE profiles. The following tables show the results of optimization (optimal geometric parameters of the cross-sectional area and optimal cross-sectional area) according to the above algorithms (Table 2 ÷ Table 6).

Table 2: Optimization results obtained by APSO algorithm

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.36	6	24.81	5.02	3.51	24.88
2	13.82	6	32.59	5.08	3.56	33.14

Table 3: Optimization results obtained by FA algorithm

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	11.79	6	21.07	5.19	3.63	25.09
2	15.87	6	28.51	5	3.5	33.30

Table 4: Optimization results obtained by CSA algorithm

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.66	6	24.12	5	3.5	24.86
2	14.39	6	31.62	5	3.5	33.08

Table 5: Optimization results obtained by GRG2 algorithm

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.66	6	23.37	5	3.5	24.48
2	14.48	6	30.63	5	3.5	32.69

Table 6: Optimization results obtained by EA algorithm

	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)
1	10.54	6	23.44	5.06	3.54	24.50
2	14.23	6	30.94	5.06	3.54	32.71

As it can be seen on the basis of the obtained results, there are unobtained whole values of the sheet metal thickness, so these values are rounded to the first higher whole value. In addition, the other values are rounded. In this way, an accurate picture of the analyzed results is obtained.

The following tables show the rounded values of the optimal geometric parameters, the optimal cross-sectional area and the achieved savings in relation to the standard profile (Table 7 ÷ Table 11):

Table 7: Optimization results obtained by APSO algorithm and achieved savings

No.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Saving (%)
1	10.4	6	24.9	6	4	27.42	17.90
2	13.9	6	32.6	6	4	36.24	21.05

Table 8: Optimization results obtained by FA algorithm and achieved savings

No.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Saving (%)
1	11.8	6	21.1	6	4	26.82	19.70
2	15.9	6	28.6	5	4	33.38	27.28

Table 9: Optimization results obtained by CSA algorithm and achieved savings

No.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Saving (%)
1	10.7	6	24.2	5	4	24.94	25.33
2	14.4	6	31.7	5	4	33.13	27.82

Table 10: Optimization results obtained by GRG2 algorithm and achieved savings

No.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Saving (%)
1	10.7	6	23.4	5	4	24.54	26.53
2	14.5	6	30.7	5	4	32.75	28.65

Table 11: Optimization results obtained by EA algorithm and achieved savings

No.	b (cm)	t (mm)	h (cm)	s (mm)	a _s (mm)	A _{opt} (cm ²)	Saving (%)
1	10.6	6	23.5	6	4	26.82	19.70
2	14.3	6	31.0	6	4	35.76	22.09

From the previous tables (Table 7 ÷ Table 11) it can be seen that the new optimal cross-sectional area values are now larger than those shown in Table 2 ÷ Table 7. In certain optimization algorithms, it is possible to introduce a constraint that the variable takes the whole value. In this case, this can be accomplished when using the Solver tool in the EXCEL software package, the GRG2, and EA optimization algorithms, using the integer option, but it is specified in a way that the results can be compared with those obtained by the algorithms in the MATLAB software package.

6.CONCLUSION

The paper presents the optimal geometric values for the welded I-beam of the pillar jib crane, for example of two jib cranes with a load capacity of 500 kg. As an object of optimization, the cross-sectional area of the cantilever is observed, where all the constraint functions are satisfied. In addition to the constraint functions, other construction and technological criteria could be used.

Optimization was carried out using the MATLAB software package by APSO, FA and CSA algorithm, as well as using the Ms EXCEL software package by GRG2 and the EA optimization algorithm. The selection of appropriate optimization methods presents their justification and achieved a saving of 28.65% (first crane) and 26.53% (second crane).

The optimization tasks have been successfully performed, as can be seen from the results in the tables (Table 2 ÷ Table 11) and shows the correctness of using the presented model for analysis and optimization.

It is seen that the best results were given by the GRG2 algorithm and the weakest by the FA (Table 2 - Table 6). In addition to the nature-inspired algorithms, the best results are achieved by CSA. In addition, it can be seen that in all cases the minimum thickness of flanges $t=6$ mm is obtained. The best results are given by GRG2, utilizing the Solver tool in the Ms EXCEL software package, or by CSA, using the MATLAB software package (Table 2 ÷ Table 6). This is equally seen from the table where the values of the results obtained are rounded (Table 7 ÷ Table 11).

The particular conclusion based on the analysis and optimization carried out obtain the fact that the welded I-beam provides significant material savings in comparison to the standard rolled I-profiles.

For intended research in this area, it is necessary to include other constraint functions that are important for analysis. In addition to the imperfections and stress

conditions in the material and welds, the elastic stability of the carrier can also be analyzed. Also, other forms of the cross-sectional area used for these types of structures, as well as other potential forms of cross-sectional area and materials that can be considered.

As a possible objective function, in further studies could be taken into account the area of the weld contour. In that case one of the multi-objective optimizations, procedures would be applied. The costs of carrier design, in addition, could be used as the goal function in the optimization process. The results obtained in this way can be verified and compared with FEM analysis, to demonstrate justification of the application of the given methods. Furthermore, in this way, some conclusions can be made as properly as a guideline for the design and optimization. Optimization can equally be done by hybrid methods, combining some of the numerical optimization procedures with FEM software packages.

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REFERENCES

- [1] R.K. Amreeta and V. Singh, "Design and Stress Analysis of Single Girder Jib Crane", International Journal of Engineering Research & Technology (IJERT), Vol. 4, Issue 09, pp. 932-936, (2015)
- [2] S.N. Khetre, P.S. Bankar and A.M. Meshram, "Design and Static Analysis of I-Section Boom for Rotary Jib Crane", International Journal of Engineering Research & Technology (IJERT), Vol. 3, Issue 8, pp. 1071-1074, (2014)
- [3] M. Dhanoosha and V.G. Reddy "Detail Design and Analysis of A Free Standing I Beam Jib Crane", International Research Journal of Engineering and Technology (IRJET), Vol. 3, Issue 12, pp. 193-203, (2016)
- [4] D. Desai and A.N. Patil, "Analysis & Optimization of Jib Crane under Influence of Various Slewing angles & load position by limit state method", International Engineering Research Journal, pp. 1207-1215

- [5] G. Pavlović, M. Gašić, M. Savković, N. Zdravković and G. Marković, “Comparative Analysis of the Model for Determination of Deflection in the Column-Mounted Jib Crane Structure”, XXIII INTERNATIONAL SCIENTIFIC CONFERENCE “TRANSPORT 2017”, Sofia (Bulgaria), 12–14 October, Vol. 15, Issue 3, article No. 1528, (2017)
- [6] S.S.Raut and B.C. Londhe, “Design and Analysis of Column Structure for Jib Crane”, International Journal for Educational Research and Interdisciplinary Studies (IJERIS), Vol. VI, Issue XI, pp. 1-6, (2017)
- [7] S.D. Kurhekar and R. Gorpude, “Design Optimization and Finite Element Analysis of Movable Jib Crane”, International Journal of Analytical, Experimental and Finite Element Analysis (IJAEFEA), Vol. 2, Issue 3, pp. 112-114, (2015)
- [8] X. Ma and X. Cheng, “Model Building of Finite Element Analysis Based on ANSYS of Column Jib Crane”, Trans Tech Publications, Vol. 462, pp. 427-433, (2012)
- [9] T. Patil and N.L. Shelke, “Structural Analysis of a Cantilever Beam with Tapered Web Section through FEA”, Journal of Steel Structures & Construction, Vol. 2, Issue 2, 1000119, (2016)
- [10] K.S. Bollimpelli and V.R. Kumar, “Design and Analysis of Column Mounted JIB Crane”, International Journal of Research in Aeronautical and Mechanical Engineering (IJRAME), Vol. 3, Issue 1, pp. 32-52, (2015)
- [11] V.S. Chavan and S.S. Waydande, “Design optimization and Modal Analysis of Cantilever I-section Beam For 0.5 ton capacity of Floor Mounted Jib Crane”, International Journal of Mechanical and Industrial Technology, Vol. 5, Issue 1, pp. 85-99, (2017)
- [12] T. Haniszewski and D. Gąska, “Numerical Modelling of I-Beam Jib Crane with Local Stresses in Wheel Supporting Flanges - Influence of Hoisting Speed”, Naše more, Vol. 64, Issue 1, pp. 7-13, (2017)
- [13] P.S. Padwal and A.R. Patil, “Review & Analysis of Cantilever Beam Crane for Weight Reduction by Using Topography Method”, International Journal of Advance Research and Innovative Ideas in Education (IJARIE), Vol. 3, Issue 4, pp. 1797-1805, (2017)
- [14] C.C. Dandavatimath and H.D. Sarode, “Finite Element Analysis and Optimization of Jib Crane Boom”, International Journal of Innovative Research in Science, Engineering and Technology (IJIRSET), Vol. 6, Issue 7, pp. 14287-14294, (2017)
- [15] K. Gandhare and V. Thute, “Design Optimization of Jib Crane Boom Using Evolutionary Algorithm”, International Journal of Scientific Engineering and Research (IJSER), Vol. 3, Issue 4, Paper ID: IJSER1553, (2015)
- [16] S. Wu and T. Su, “Optimization Design of Cantilever Beam for Cantilever Crane Based on Improved GA”, TELKOMNIKA Indonesian Journal of Electrical Engineering, Vol. 12, No. 4, pp. 2652-2657, (2014)
- [17] G. Pavlović, A. Stepanović, J. Vidaković, M. Savković and N. Zdravković, “Design Optimization of the Box Section of the Single-Girder Bridge Crane by Generalized Reduced Gradient Algorithm”, IMK-14, Research & Development in Heavy Machinery, Vol. 21, No. 3, pp. EN79-EN86, (2015)
- [18] G. Pavlović, M. Gašić, M. Savković and N. Zdravković, “Comparative Analysis of Local and Lateral Stability of Plates as the Constraint Functions Within Optimization of Main Girder Box Section of the Bridge Crane”, IMK-14 – Research & Development, Vol. 18, No. 1, pp. EN11-EN18, (2012)
- [19] <https://www.mathworks.com/matlabcentral/fileexchange/29725-accelerated-particle-swarm-optimization>
- [20] https://www.mathworks.com/matlabcentral/fileexchange/29693-firefly-algorithm?s_tid=srchtitle
- [21] <https://www.mathworks.com/matlabcentral/fileexchange/29809-cuckoo-search-cs-algorithm>
- [22] D. Ostrić and S. Tošić, “Dizalice”, Institut za mehanizaciju Mašinskog fakulteta Univerziteta u Beogradu, Beograd, (Srbija), (2005)
- [23] Z. Petković and D. Ostrić, “Metalne konstrukcije u mašinogradnji I”, Institut za mehanizaciju Mašinskog Fakulteta Univerziteta u Beogradu (Srbija), (1996)