Simulacija tipičnog incidentnog događaja kod okvirne noseće strukture

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U nedostatku eksperimentalnih analiza vibracija, mogu se uspešno vršiti numeričke analize nosećih struktura, primenom savremenih softvera za dinamiku mašina. Projektovanje u slučaju maksimalnih opterećenja ima za cilj da očuva stabilnost strukture, pa se u tom cilju obavljaju vrlo različite simulacije procesa u toku opterećenja. Osnovni odgovori o preraspodeli unutrašnjih sila u strukturi izazvani naglim događajima mogu se tražiti dinamičkom analizom prelaznih stanja – tranzijentnom dinamikom. U ovom radu je pokazan postupak simulacije dinamičkog ponašanja toranjske dizalice u toku udarnog radnog opterećenja. U radu je takodje pokazan i razvijen mehanički model strukture kao i FEM analiza – simulacija ponašanja toranjske dizalice u odabranom ekstremnom režimu rada. Numeričkom analizom pokazan je postupak određivanja nekih najvažnijih podataka za istraživače i konstruktore nastao incidentom, kao što su: najveći napon, zakon kretanja (oscilovanja) i unutrašnje sile strukture. Istraživanja se oslanjaju na rezultate aktuelnih merenja. Numerički je testirana dinamička stabilnost strukture u tranzijentnom režimu sa opterećenjem koje je u granicama maksimalnog opterećenja dizalice. Time je istraživana stabilnosti visokog objekta izloženog dinamici u uslovima opasnih uticaja. Poznavanje ove dinamike ima za cilj monitoring i kontrolu dinamičkih sila u strukturi.

Keywords: Simulacija opasnih uticaja, tranzijentna analiza, rešetkasta struktura, toranjska dizalica

1. UVOD

Dinamičko opterećenje struktura često u praksi ima karakter udara. Poseban vid udarnog dejstva je impuls. Impulsna funkcija se odlikuje silom visokog intenziteta u kratkom vremenskom periodu. Impulsno dinamičko opterećenje posebno može ugroziti stabilnost visokih rešetkastih struktura [1]. Kod nekih struktura značajni uticaji se javljaju čak i pri ekstremnim radnim dejstvima: Kod toranjskih dizalica, velika brzina dizanja ili spuštanja tereta i naglo kočenje uvodi izrazite impulsne uticaje od inercije pokretnih masa. U toku preraspodele opterećenja rešetkaste strukture ispoljavaju osobinu – rezerviranja [2]. To je sposobnost strukture za preuzimanje unutrašnjih sila od okolnih elemenata u zoni otkaza ili preopterećenja. Sposobnost rezerviranja strukture se može proveriti simulacijom incidenta. U tu shvrhu moraju se odrediti potencijalni scenariji ekstremnih situacija. Više izvedenih simulacija može uticati da se popravi bazni statički dizajn strukture [3,4]. Danas se za simulacije iznenadnih otkaza koriste metode dinamičkih analiza nestacionarnih kinematskih stanja, poznate kao tranzijentne analize [5]. Jedna takva analiza, koja ima za cilj proveru rezerviranja, pokazana je ovim radom.

Istraživanja su obavljena na široko zastupljenoj klasi toranjskih dizalica [6, 7]. Izabrana je tipična rešetkasta struktura i geometrija kakvu ima dizalica Potain 744e. Cilj istraživanja je modeliranje incidenta izazvanog opasnim (udarnim) uslovima rada. U tu svrhu razvijen je mehanički model i postupak provere kritičnih (lokalnih) članova strukture na dinamičko dejstvo predpostavljenog globalnog uticaja [8]. Za simulaciju je posmatrana najčešća kinematika – operacija udarnog dizanja tereta.

2. TEORIJSKA OSNOVA DINAMIČKE ANALIZE

Karakteristike dinamičkog procesa definišu uslovi (granice) koji se primenjuju u daljem formiranju i smanjenju dinamičke strukturne jednačine. Osno

definisana modela dinamičke strukture i uslova daje matrica krutosti.

Prvi uslov (predpostavka) linearne analize podrazumeva ograničena pomeranja strukture - mala u odnosu na dimenzije modelirane strukture. Druga predpostavka linearne analize je čisto elastično ponašanje materijala. Treća pretpostavka linearnih analiza je nepromenljivost graničnih uslova za vreme dejstva opterećenja. Prva pretpostavka omogućuje zanemarivanje promena u samoj strukturnoj konfiguraciji. Time matrica krutosti $[K]$ zadržava istu formu (vrednost) pa se rešenja dobijaju integracijom originalne strukture. Unutrašnje elastične strukturne sile u članovima dobijaju se direktnim načinom, množenjem matrice krutosti i izračunatih pomeranja. Druga pretpostavka omogućuje korišćenje konstantne matrice materijala. Opšti oblik dinamičke jednačine oscilovanja strukture sa m članova pod dejstvom eksterne poremećajne sile predstavlja se matričnom formom (1):

$$[a] \cdot \{\ddot{q}\} + [b] \cdot \{\dot{q}\} + [k] \cdot \{q\} = \{F(t)\} \quad (1)$$

Gde je:

$[a]$ - matrica inercijalnih koeficijenata je označena,

$[b]$ - matrica viskoznog prigušenja,

$[k]$ - matrica krutosti strukture;

$\{q_i\}$ - vektori generalisanih koordinata za $(i=1,...,m)$,

$\{\dot{q}\}, \{\ddot{q}\}$ - izvodi po vremenu,

$\{F(t)\}$ - generisana sila.

Rešenje sistema jednačina (1) je vektor generalisanih pomeranja $\{q_i\}$ kao odgovor strukture na spoljnu dinamičku pobudu. Rešenje se dobija numerički, direktnom integracijom jednačina strukture primenom Newmarkove, Wilsonove ili druge metode integracije. Analiza uključuje ukupno strukturno prigušenje G , proporcionalno dominantnim susednim kružnim frekvencijama ω^* i ω^{**} (3). Strukturno prigušenje potiče iz prigušenja u elementima i vezama [10]. Prigušenje se

zato izražava preko matrice krutosti $[k]$ u kojoj figurišu $[k_G]$ – globalna matrica krutosti i $[k_E]$ – matrica krutosti elemenata, odnosno:

$$[k] = (I + i \cdot G) \cdot [k_G] + i \cdot \sum G_E [k_E] \quad (2)$$

Gde je:

$[k_G]$ – opšta matrica krutosti strukture,

$[k_E]$ – matrica krutosti elemenata,

G – strukturno prigušenje,

G_E – strukturno prigušenje elemenata,

i – imaginarni broj.

Kako tranzijentna analiza ne obuhvata proračun preko kompleksnih brojeva treba izvršiti konverziju strukturnog prigušenja u ekvivalentno viskozno prigušenje. Konverzija se vrši jednačinom (3) definisanjem kompleksne matrice prigušenja $[b_C]$, pomoću dominantnih kružnih frekvencija iz prethodno izvršene modalne analize:

$$[b_C] = [b] + \frac{G}{\omega^*} [k_G] + \frac{I}{\omega^{**}} \sum G_E [k_E] \quad (3)$$

Gde je: ω^* i ω^{**} – kružne frekvencije.

Ovaj rad razmatra slučaj uključivanja opasnih situacija u strukturno ponašanje konstrukcije. Opasna situacija se inicira opterećivanjem toranjske dizalice tegom od 2,2 t na njenom najvećem dohvatu od 45 m usled čega dolazi do elastičnih deformacija strukture. Istraživanjima [8] izvedenim na toranjskoj strukturi utvrđena je razlika pomeranja linearnog i nelinearnog modela statičke analize, ispod 1.054 %. Prihvatajući ove razlike kao ograničene, dalja analiza je izvedena linearnim metodama.

Vrednosti frekvencija ω^* i ω^{**} predstavljaju frekvencije koje odgovaraju vrstama tranzijentne analize. U ovoj studiji slučaj vertikalne oscilacije strukture su dominantna. Za izabrane oblike modova stanja najniže frekvence ω^* i ω^{**} izabrane su iz tabele 1 i tabele 2, respektivno.

Prvi korak analize modela odnosi se na određivanje sopstvenih frekvencija i sopstvenih oblika oscilovanja. To je modalna analiza. Dinamička jednačina slobodnog neprigušenog oscilovanja izražena u matricnoj notaciji (4), za linearni sistem, ima harmonijsku formu, (5):

$$[M] \cdot \{\ddot{q}\} + [K] \cdot \{q\} = \{0\} \quad (4)$$

Gde je:

$[M]$ – matrica masa,

$[K]$ – matrica krutosti strukture,

$\{q\}$ i $\{\ddot{q}\}$ – vektori sopstvenih oblika oscilovanja.

$$\{q\} = \{\Phi\}_i \cdot \cos \omega_i \cdot t \quad (5)$$

Gde je:

ω_i – i^{th} prirodna cirkularna frekvencija,

$\{\Phi\}_i$ – sopstveni vektor za i -tu cirkularnu frkfencu,

t – vreme.

Rešenje sopstvenih (prirodnih) neprigušenih frekvencija i oblika (modova) strukture, nalazi se iz frekventne jednačine (6):

$$[K] - \omega^2 \cdot [M] \cdot \{\Phi\} = \{0\} \quad (6)$$

3. MODELIRANJE OKVIRNIH STRUKTURA

Rešetkaste i okvirne strukture, obzirom na složenost i broj članova, modeliraju se diskretnim modelima sa konačnim brojem stepeni slobode (slike 1 i 2), [15,16]. Da bi stepen aproksimacije bio nizak, razvijaju se modeli visoke vernosti, za šta se koristi *Finite Element* struktura. Štapovi i profili od kojih je sačinjena struktura mogu se predstaviti linijskih konačnih elemenata tipa BEAM i ROD. Osnova formiranja modela je *grid* struktura izrađena na osnovu sistemnih i težišnih statičkih linija fizičkog objekta. Grid struktura je struktura čvorova u kojima su povezani konačni elementi. U čvorovima su statička nepoznata pomeranja (određena vezama članova), poznata u metodi deformacija kao *Degree of Freedom* (DOF). Kompaktni elementi značajnih masa predstavljaju se tačkastim konačnim elementima masa. Fundament izveden od armiranog betona, predstavlja se solid 3D konačnim elementima, a elastično oslanjanje na tlu modelira se opružnim (spring) linijskim konačnim elementima (SI.2).

Figure 1: FE model diskretnih lineranih struktura (strelna platforma, veza, platforms). Struktura ima 1673 konačna elementa i 1141 čvora

Figure 2: FE Model oslanjanja tornja dizalice

Slika 3-A: Osnovni oblici oscilovanja strukture u vertikalnoj ravni (Mod-2). Masa $m_Q=2200\text{kg}$, podignuta visoko do strele.

Slika 3-B: Osnovni oblici oscilovanja strukture u vertikalnoj ravni (Mod-6). Masa $m_Q=2200\text{kg}$, podignuta visoko do strele.

Slika 3-C: Osnovni oblici oscilovanja strukture u vertikalnoj ravni (Mod-10). Masa $m_Q=2200\text{kg}$, podignuta visoko do strele.

4. MODALNA ANALIZA

Oblik oscilovanja konstrukcije – $\{\Phi\}_i$ – sopstveni vektor (oblik moda) za osnovne kružne frekvence prikazan je na slikama 3-A, 3-B i 3-C. Frekvence oscilovanja moda iznosi $\omega_2 = 0,3756$. Modalna analiza (*Normal modes analysis*) radi se radi definisanja izraza (3) kojim se opisuju uslovi prigušenja sistema. Naime kod prigušenih oscilacija nosećih struktura, predpostavlja se da su otporne sile proporcionalne brzini oscilovanja. Zato se moraju da izdvoje sopstvene frekvencije bliske frekvencijama prinudnog ili slobodnog oscilovanja. To je prvi korak u tranzijentnu i frekventnu analizu mehaničkog sistema.

Za neposrednu realizaciju modalne analize ove strukture korišćen je *Lanczos* method, modifikovan *Givens* method i modifikovan inverzni power method *Sturm-a* za identifikovanje najnižih vrednosti, [11].

Naredna tabela 1, daje pregled rezultata sopstvenih frekvencija, značajnih za oscilovanje tornja u vertikalnoj ravni, bez tereta. Tabela 2 koja sledi daje pregled značajnih frekvencija oscilovanja strukture u vertikalnoj ravni, sa teretom $m_Q = 2200$ kg, visoko podignutog do same strele.

Table 1 Frekvenca oscilovanja toranjske dizalice bez tereta

$m_Q = 0$ kg	Mod-2	Mod-4	Mod-6	Mod-10
ω Hz	0,45719	1,5380	2,0042	3,3565

Table 2 Frekvenca oscilovanja toranjske dizalice sa teretom

$m_Q = 2200$ [kg]	Mod-2	Mod-4	Mod-6	Mod-10
ω [Hz]	0,37565	1,3295	1,6009	3,1499

5. SIMULACIJA UDARNOG INCIDENTA

Spoljašnji impulsni uticaji se mogu saopštiti diskretnoj strukturi silama $F(t)$, koje su funkcije kratkih vremenskih intervala. Izbor oblika poremećajne sile izvršen je na osnovu eksperimentalnog izučavanja razvoja događaja u realnoj strukturi pri naglim dejstvima [8]. Za poremećaj je iskustveno (prema tim istraživanjima) izabran jedan oblik normirane poremećajne sile $\Delta F/F$, prikazan na slici 4. Forma poremećajne sile koristi tri karakteristična vremena simulacije: Δt_{BL} vreme simulacije pre podizanja tereta (*before lifting time*), Δt_L vreme rasta sile usled dizanja tereta i vreme posmatranja ponašanja posle dizanja tereta: Δt_{AL} (*after lifting*). Teret mase 2200 kg je koncentrisano raspoređen na četiri tačke strele (4x550 kg).

Funkcija poremećajne (pobudne) sile $F(t)$ je impulsnog tipa. Definiše se na bazi ekstremnog tereta mase $m = 2200$ kg, i ekstremnog dohvata strele $L = 45$ m. Vreme $\Delta t_{BL} = 40$ (s), je procenom izabrano, tako da odgovara potrebnom vremenu smirenja oscilovanja sopstvenih masa strukture. Tada struktura zauzima položaj statičke ravnoteže. Nakon smirenja dolazi do podizanja tereta, brzinom od približno $v = 10$ m/min. Ovako brzo, naizgled nekontrolisano podizanje, može biti posledica nepažnje rukovaoca. Ovo je ekstremna regularna situacija sa teretom koja produkuje izrazitu dinamiku bez posledica po stabilnost. Ta situacija je eksperimentalno isprobana. Ovako formiran poremećajni uticaj uveden je u čvorove

strukture na mestima vezivanja užadi za strukturu. Dinamički model strukture, na slici 1 i 2, razvijen je primenom softvera MSC [9].

Slika 4: Rastuća ramp-funkcija poremećajne sile F_{E-158}

6. TRANZIJENTNA ANALIZA

Tranzijentna analiza spada u tranzijentnu dinamiku što odgovara prelaznim kinematskim stanjima članova strukture. Tranzijentna dinamika izazvana je uticajem promenljivog neperiodičnog dejstva na posmatrani objekat. U ovom primeru posmatrani objekat je toranjska dizalica koja se opterećuje udarno, podizanjem tereta, mase 2200 kg u najvećem dohvat u od 45m. Koeficijent prigušenja strukture je $G = 0,04$. Periodični nivo ontegracije je $\Delta t = 0,01$ s. Kritična frekvenca prigušenja je $\omega_2 = 0,37565$ Hz i $\omega_4 = 1,6009$ Hz.

Rešenja tranzijentne analize su tražena u dva odgovorna elementa FE skupa. To su vertikalni pojasni elementi tornja (prednji pojasni nosač tornja E-1656 i zadnji pojasni nosač tornja E-229), sl.1. Posebno je izloženo oscilovanje vrha strele pri udaru (Node 1410). Analiza se vrši pomoću [11], iskustvo sa teorijskim modeliranjem [12], [14] i definisanim prigušenjem [13].

6.1. Analiza prednjeg pojasnog profila tornja

Maksimalni uzdužni komponentni napon prednjeg pojasnog profila tornja (Element-1656 u modelu, sl.1) određen ovom analizom, javlja se u trenutku podizanja tereta mase 2200 kg pri najvećem dohvat u. U režimu mirovanja pre podizanja, statički uzdužni komponentni napon iznosi $38,5E \cdot 10^6$ N/m². Nakon podizanja tereta, impulsnom silom, simulacijom je dobijen ekstremni napon od $-69,5E \cdot 10^6$ N/m², slika 5. Ovaj dijagram pokazuje da unutrašnja sila menja pravac, izazivajući pritisak

Slika 5 Komponentni uzdužni naponi pri podizanju tereta u E-1656 (prednji pojasni član tornja). Jedinice: napon N/m², vreme (sec.) Promena srednjeg komponentnog napona pri podizanju je $56,5106$ N/m²

6.2. Analiza zadnjeg pojasnog profila tornja

Najveća pomeranja su na kraju strele toranjске dizalice, pa je posebno, simulacijom posmatrano pomeranje vrha strele. Na slici 7 prikazani su rezultati simulacije od impulsnog podizanja tereta. Dobijeno je vertikalno stacionarno spuštanje vrha strele za 0.6565 (m) od podizanja tereta mase 2200 kg.

Figure 6: Pomeranje vrha strele pri podizanju tereta od 2200 kg u maksimalnom dohvatu od 45 m.

Opseg maksimalne promene visine oscilovanja vrha strele pri impulsnom udaru je blizu 1.18 m (gledano iz poslednje amplitude pre udara). Prethodne amplitude, usled slučajnih ili slobodnih oscilovanja, sabrane sa pomeranjima izazvanim incidentnim udarom, mogu da prekorače ukupna vertikalna pomeranja vrha strele od 1.5 m. Ovako značajna pomeranja čine oko 3,33 % od dužine strele. Taj procenat ukazuje da je poželjno uraditi nelinearnu analizu za viši kvalitet istraživanja.

7. EKSPERIMENTALNA PROVERA

Merenjem, pokazanim dalje na slici 7, izloženi su rezultati kontrolnog eksperimentalnog ispitivanja sa teretom od $m_Q=2200$ kg. Eksperiment je urađen tenzometrijskim putem, primenom mernog pojačivača HBM MGC+, korišćenjem mernih senzora HBM LY 11/120. Paralelno je izmeren ugib vrha strele, direktnim zapisom na papiru. Eksperiment je detaljnije opisan u radu [8]. Zadovoljavajuća tačnost je postignuta u frekventnom domenu, poređenjem perioda oscilovanja dobijenog simulacijom i eksperimentom prikazanim na slici 8. Eksperiment: Merenje br.8: $m_Q=0$ kg, deset oscilacija je dobijeno za vreme od 22.62 s: $10 \cdot T_{EXP} = 176.77-154.15 = 22.62$ s, $T_{EXP}=2.262$ s. Period oscilovanja iz modalne analize je određen: ($m_Q=0$ kg), $T_{MOD}=2.18726$ s, Tabela I.

Slika 8: Eksperimentalno ispitivanje impulsnim režimom: Mikrodeformacije pojasnih nosača tornja strukture MT1, MT2. Masa pokude $m_Q=2200$ kg, Ugao usmerenja strele 65° , Prosečna perioda $T_0=2,262$ s

Relativno odstupanje ova dva perioda je dato izrazom (7) ε_T :

$$\varepsilon_T = \frac{T_{EXP} - T_{MOD}}{T_{EXP}} \cdot 100 = \frac{2,262 - 2,18726}{2,262} \cdot 100 = 3,304\% \quad (7)$$

gde je:

T_{EXP} - eksperimentalna perioda oscilovanja,

T_{MOD} - perioda oscilovanja modalne analize.

Relativna razlika vertikalnih translacija vrha strele, eksperimenta i simulacije ($m_Q=2200$ kg) je data izrazom (8) ε_Z :

$$\varepsilon_Z = \frac{Z_{EXP} - Z_{SIM}}{Z_{EXP}} \cdot 100 = \frac{0,68 - 0,6565}{0,680} \cdot 100 = +3,456\% \quad (8)$$

gde je:

Z_{EXP} - eksperimentalna vrednost vertikalne translacije kraja strele,

Z_{SIM} - simulirana vrednost vertikalne translacije kraja strele.

Poređenjem dijagrama simulacije sl.5 sa dijagramom impulsnog podizanja tereta datog na slici 8, očigledna je bliskost teorijskih i eksperimentalnih rezultata. Male razlike koje se mogu uočiti potiču od različitog položaja usmerenosti strele. Dijagram na slici 8 pokazuje izmerene deformacije pri podizanju tereta u korenu tornja strukture na mernoj traci 1 (MT1) i mernoj traci 2 (MT2). U merenju je impuls pri dizanju tereta bio nešto duži nego pri simulaciji (oko 3 s). Dobijeni rezultati simulacijom po kvalitetu i veličini, odgovaraju ponašanju realne strukture. Slika 9 pokazuje fotografije objekta i merne opreme.

Figure 7: Deformacije određene eksperimentalno na tornju strukture MT1. Slobodno oscilovanje tornja po spuštanju tereta (merenje 8)

Slika 9 Fotografije sa eksperimentalnog testiranja toranjske dizalice na impulsni režim i merna oprema

8. ZAKLJUČAK

Izložen koncept simulacije je realnost savremene računarske i praktične tehnologije. Procedura se može uspešno primeniti na odgovorne strukture izložene specijalnim dejstvima, posebno incidentnim. Pri tome se može zaključiti:

Kvalitet simulacije uvodi mogućnost potpune kontrole rezerviranja strukture pre svega zbog više hiljada članova u okvirnim i rešetkastim strukturama. Taj obim numerike se klasičnim projektovanjem nije mogao dostići.

Ovako opisanim tranzijentnim modelom redefinisan je statički model dizajna struktura. Uveden je sistem dinamičkog testiranja na specijalne uticaje kojim se garantuje pouzdanost na bazi ciljnog i zadatog.

Ovako kompleksno sagledana struktura definiše kritične tačke članova i kritična dejstva. Iz tih informacija se može onda razvijati monitoring. Izbor lokacija za monitoring treba izvršiti na članovima sa najnižim nivoom rezerviranja.

Posmatrana struktura toranjske dizalice se odlikuje visokim nivoom amplituda oscilovanja, što je teško prihvatljiva kategorija za prirodu čoveka. Poboljšanje je moguće povećanjem geometrije preseka tornja, strela i fundamenta.

Dobijena pomeranja strukture veća od 3% geometrije upućuju da je poželjno raditi nelinearne tranzijentne analize za bolji kvalitet istraživanja i tehničkog dizajna što je nova opcija savremenih softvera.

ZAHVALNOST

Rad predstavlja deo istraživanja u okviru projekta TR 35049, Mašinski fakultet Univerziteta u Nišu. Autori

se zahvaljuju Ministarstvu prosvete, nauke i tehnološkog razvoja Republike Srbije.

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Typical Incident Simulation of Frame Supporting Structures

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In the absence of experimental vibration analysis, numerical analysis of supporting structures can be successfully performed by applying the software for machine dynamics. The design in the case of maximum stress aims to maintain the stability of the structure and, for that purpose, very different simulations of process are performed. This paper presents the dynamic simulation during extreme load. A mechanical model of the structures is developed and the Finite Element Analysis (FEA) of the tower crane is performed. Numerical analysis shows a procedure to determine some important data for researchers acquired from tower crane incidents such as: the highest stress, law of deflection (structure vibrations) and internal forces of structure. The theoretical (numerical) model is verified by using the experimental (natural) model. From these analyses, critical impacts and critical behavior of the structure are determined.

Keywords: Dynamic response, critical impacts, transient analysis, tower crane.

1. INTRODUCTION

Research in the area of machine dynamics is the subject of many papers. The paper [1] researches elevator dynamics from the aspect of defining the law of mass movement. The crash (damage) has the special importance as a direct consequence of mechanical system dynamics [2]. In that case, the dynamic load structure often has an impulse character.

There is a special form of impact load, named impulse. Impulse function is characterized by the force of high intensity in a short-time period. Impulse dynamic stress can have a particular impact on the stability of frame structures [3]. Some structures are affected during extreme operating actions: high speed of lifting or lowering load or sudden brake leads to a certain impulse impact from the inertia of moving mass [4]. At high velocities and great heights we observe vertical vibrations of the dynamic model with a finite number of degrees of freedom.

During stress reallocation, frame structures impose the character of reserving [5]. It is the structural ability for redistribution of internal forces on the tower crane elements in the failure zone. The ability of structure reserving can be tested by incident simulation. Multi-performed simulations can have an impact on improving the basic static design of a structure [5]. Today, the methods of dynamic analysis of non-stationary kinematic states are used for simulations of sudden failure, known as transient analysis. In this paper, a potential scenario of extreme situations is determined. These analyses intend to determine the reserve capacity of members of the structure.

Experimental research has been performed on a typical class of tower cranes [6, 7]. Typical frame structure and geometry is chosen, the same as in Potain 744e crane type, Fig. 1. The aim of this paper is conduct an incident simulation caused by critical load impact during operating actions. Development and verification of the mechanical model structure is performed within research [8]. For that reason, a mechanical model was developed and load impact on critical (local) elements of the structure were analyzed. The most frequent kinematic, lifting operation impact on the tower crane was observed for simulation.

2. THEORETICAL BASIS OF DYNAMIC ANALYSIS

Characteristics of a dynamic process define conditions (limits) which are applied in further forming and reducing of the dynamic structure equation. The basis of the model structure description and boundary conditions are given by the stiffness matrix.

The first constraint of linear analysis considers limited structure movement – small in relation to dimensions of a modeled structure. The second constraint of linear analysis is pure elastic behavior of materials. The third constraint of linear analysis is the invariability of boundary constraints during the stress effect. The first constraint allows the analyst to neglect the change in structural configuration. As a consequence, the stiffness matrix $[k]$ is evaluated by performing the integration over the original structural configuration. The elastic (internal) structural forces are obtained in a straight-forward manner by multiplying the stiffness matrix by displacements. The second assumption allows the usage of a constant constitutive matrix of the material. The general form of the dynamic equation for structure vibrations with element m under the influence of the external deviation force is given in the following matrix form:

$$[a] \cdot \{\ddot{q}\} + [b] \cdot \{\dot{q}\} + [k] \cdot \{q\} = \{F(t)\} \quad (1)$$

where:

$[a]$ - matrix of inertial coefficients,

$[b]$ - matrix of damping,

$[k]$ - matrix of structure stiffness;

$\{q_i\}$ - vectors of generalized coordinates for $(i=1,...,m)$,

$\{\ddot{q}\}, \{\dot{q}\}$ - derivatives per time,

$\{F(t)\}$ - generalized forces.

The solution of the equation system (1) is the vector of generalized displacements (motion) $\{q_i\}$ as the response of the structure to the external dynamic impulse. The solution is obtained numerically, by direct integration of the equation structure and application of *Newmark*, *Wilson* or some other method of integration. The analysis includes the overall structural damping G [8], proportional to the dominant neighbor circular frequencies ω^* and ω^{**} (eq. 3). Structural damping comes from damping in

elements and relations [10]. For that reason, damping is expressed through the stiffness matrix $[k]$ and it can be obtained in the form:

$$[k] = (I + i \cdot G) \cdot [k_G] + i \cdot \sum G_E [k_E] \quad (2)$$

where:

$[k_G]$ – global stiffness matrix,

$[k_E]$ – stiffness matrix of elements,

G – overall structural damping,

G_E – structural damping of elements,

i – imaginary number.

As transient analysis does not consider calculation via complex numbers, it is necessary to carry out the conversion of structural damping into equivalent-viscous damping. The conversion is obtained by equation (3), by defining the complex matrix of damping $[b_C]$, and by the dominant circle frequencies from the previously performed model analysis:

$$[b_C] = [b] + \frac{G}{\omega^*} [k_G] + \frac{I}{\omega^{**}} \sum G_E [k_E] \quad (3)$$

where: ω^* and ω^{**} – circular frequencies.

This paper considers the case of including the critical situation into the structural behavior of construction. The critical situation is initiated by loading the tower crane with 2200 kg heavy load on the highest reach of 45 m where elastic translations of the structure occur. Previous research on tower structures [8] shows the difference in the results of static analyses of the linear and non-linear model which is below 1,054 %. Accepting these differences as limited, further analysis is performed by using linear methods.

The value of frequencies ω^* and ω^{**} represent the frequencies that correspond to the type of transient analysis. In this case study the vibratory states of structure in the vertical global plane are dominant. The mode shapes which these states describe by the lowest frequency ω^* as well as ω^{**} are selected from next tabel 1 and table 2.

The first step of modal analysis is related to determining natural frequencies and natural forms of vibrations. That is the modal analysis. The dynamic equation of the free undamped vibration expressed in the matrix notation (eq. 4), for a linear system, has the harmonic form (eq. 5):

$$[M] \cdot \{\ddot{q}\} + [K] \cdot \{q\} = \{0\} \quad (4)$$

where:

$[M]$ – overall inertial mass matrix,

$[K]$ – global stiffness matrix,

$\{q\}$ – vector of translations,

$\{\ddot{q}\}$ – vector of accelerations.

$$\{q\} = \{\Phi\}_i \cdot \cos \omega_i \cdot t \quad (5)$$

$$[K] - \omega^2 \cdot [M] \cdot \{\Phi\} = \{0\} \quad (6)$$

where:

ω_i – i^{th} circular natural frequency in radians per unit time,

$\{\Phi\}_i$ – eigenvector (mode shape) for i^{th} circular frequency,

t – time.

After replacing the solution $\{q\}$ and $\{\ddot{q}\}$ (eq. 5) in the system of equations (4), the characteristic matrix form

eigenequation (eq. 6) is obtained, from which eigenvalues and mode shapes of system are obtained.

3. GRID FRAME STRUCTURE MODELING

Considering the large and various numbers of frame elements, the modeling of tower cranes is very complex. For that reason, frame structures are modeled by discrete models with a finite number of degrees of freedom (fig. 1 and 2), [15,16]. In order to keep the low level of approximation, high-valued models are developed by using the Finite Element (FE) structure. The frame structure can be illustrated by linear finite elements of *beam* and *rod* type. The base of model forming is the structure, designed on the basis of the system and static medians of a physical object.

Grid structure is a structure of nodes in which finite elements are connected. Nodes are filled with unknown static displacements (determined by element correlation), known in the deformation method as Degrees of Freedom (DOF). Compact elements of significant masses are illustrated by dotted finite mass elements. The base, built of reinforced concrete, is illustrated by solid 3D finite elements. The elastic support on the ground is modeled by linear spring finite elements (fig. 2).

Figure 1: FE model of discrete linear structures (boom tower cranes, tie, platforms). The structure has 1673 finite elements and 1141 nodes

Figure 2: FE Model of base and frame structure of tower crane

Figure 3-A: Basic eigenforms of structure vibrations on the vertical surface (Mod-2). Mass $m_Q = 2200$ kg, lifted to the boom.

Figure 3-B: Eigenforms of structure vibrations on the vertical surface (Mod-6). Mass $m_Q = 2200$ kg, lifted to the boom.

Figure 3-C: Eigenforms of structure vibrations on the vertical surface (Mod-10). Mass $m_Q = 2200$ kg, lifted to the boom.

4. MODAL ANALYSIS

The form of oscillating structure – $\{\Phi\}_i$ – eigenvector (mode shape) for *basic* circular frequency is illustrated in Fig. 3-A, fig. 3-B and fig. 3-C. The modal form of vibrating frequency $\omega_2 = 0,3756$ Hz is given. The *Normal mode analysis* is performed to define the expression (eq. 3) by which the conditions of the damping system are described. It is assumed that damped vibrations of the support structure are resistant forces proportional to the speed of vibrations. That is why natural frequencies, closer to the frequencies of forced or free vibrations, must be separated. This is the first step in the transient and frequent analysis of a mechanical system. For immediate realization of modal analysis for this structure, the *Lanczos* Method, the modified *Givens* method, and the modified inverse power *Sturm* method are used for identification of lowest values [11].

Table 1 shows the preview of natural frequencies, important for the oscillation of the tower crane on the vertical plane, **without load**. Table 2 shows frequencies of structure oscillations on the vertical plane, **with load** $m_Q = 2200$ kg lifted high to the boom.

Table 1 Natural frequencies of the tower crane without load

$m_Q = 0$ kg	Mod-2	Mod-4	Mod-6	Mod-10
ω Hz	0,45719	1,5380	2,0042	3,3565

Table 2 Natural frequencies of the tower crane with load

$m_Q = 2200$ [kg]	Mod-2	Mod-4	Mod-6	Mod-10
ω [Hz]	0,37565	1,3295	1,6009	3,1499

5. SIMULATION WITH IMPACT LOAD INCIDENT

External impulses can be attached to discrete structure by forces $F(t)$, which are the functions of short-time intervals. The selection of the excitation force form is done according to the experimental studying of event in a real (physical) structure due to impulsive effects [8]. For excitation, one shape of unit excitation force $\Delta F/F$, illustrated in Fig. 4, is chosen. The form of excitation force uses three characteristic periods of simulation: Δt_B , the time of simulation before lifting the load (before lifting time), Δt_L , the time of growing force due to lifting the load (experimentally quantified), and the time of behavior observed after lifting the load: Δt_{AL} (after lifting). The load of 2200 kg is concentrated at the end of crane boom (4x550 kg).

The function of excitation force $F(t)$ has the character of the impulse type. It is defined on the basis of extreme load mass $m = 2200$ kg and extreme crane boom reach $L = 45$ m. Time $\Delta t_{BL} = 40$ s is chosen by evaluation, thus it corresponds to necessary time for the conciliation of the vibrating natural structure mass. Then, the structure occupies the position of static balance. After the conciliation of the structure, there is the lifting load, speeding approximately $v = 0,16$ m/s. Such a fast, uncontrollable load lifting could be a consequence of a careless operator of the crane. This is an extreme regular situation with a load which produces particular dynamics without consequences on the stability. This situation is experimentally tested. Thus formed excitation impact of

load is placed into the nodes of the structure where ropes are tied to the structure model. Software: MSC Nastran 2004 [9].

Figure 4 Growing ramp-function excitation force F_{E-158}

6. TRANSIENT ANALYSIS

Transient analysis belongs to transient dynamics, which corresponds to transitive states of structure elements. Transient dynamics is caused by the impact of a variable non-periodic effect on the observed object. This example shows that the observed object is a tower crane which is stressed by impact load, by lifting 2200 kg of load, at the highest reach of 45 m. The damping structure coefficient is $G = 0,04$. The periodic level of integration is $\Delta t = 0,01$ s. Critical damping frequencies are: $\omega_2 = 0,37565$ Hz and $\omega_4 = 1,6009$ Hz.

Solutions of transient analysis are found in two responsible elements of the FE group. These are vertical elements of the tower crane (the front tower crane beam E-1656 and the back tower crane beam E-229), Fig. 1. Vibration of the end of the boom (Node 1410) during the impact load is also specially analyzed. The analysis is performed by using the procedures [11], experience with theoretical modeling [12], [14] and dumping defined [13].

6.1. Analysis of the front tower crane beam

Maximum longitudinal component stress of the front crane tower beam (E-1656 in model, Fig. 1) defined by this analysis, occurs at the moment when the lifting load of 2200 kg is at the highest reach. During the inactivity of the tower crane and before lifting, the longitude component stress is $38,5E \cdot 106$ N/m². After lifting the load by the impulsive force, the extreme stress is obtained by simulation and it is $69,0E \cdot 106$ N/m², Fig. 5. This diagram shows that the internal force changes direction, causing stress.

Figure 5: Component longitude stress during the lifting of the load in E-1656 (front belt tower crane element). Units: stress N/m², time s. Change of middle dynamic component stress during lifting is $56,510^6$ N/m².

6.2. Analysis of moving the end of crane boom

The biggest translation (local motion - displacement) is at the end of the crane boom, thus by simulation, the translation of the end boom is specially observed. Fig. 6 illustrates the results of the impulsive load lifting simulation. The vertical stationary lowering of the end boom is obtained for 0.6565 m from lifting the load of 2200 kg.

Figure 6: Translation of the end of the boom Regime: 2200 kg load lifting on the maximal reach of 45 m.

The range of the maximum vertical translation caused by the impulsive impact load is close to 1.18 m (from the last amplitude, before the impact load). Previous amplitudes are summed up with all translation caused by the incidental impact load, due to accidental or free vibrations, and they can exceed the complete vertical end boom translation of 1.5 m. Such significant translation consists of 3.33 % boom length. That percentage shows that it is necessary to perform the non-linear analysis to achieve a higher quality of research.

7. EXPERIMENTAL VERIFICATION

Measures, in Fig. 7, show the results of the control experimental research with $m_Q = 2200$ kg load. The experiment was done in the tensometrical way, by using an HBM MGC+ measure amplifier and HBM LY 11/120 measuring sensors (gauges). The deflection of the end of the boom is measured parallelly. The experiment is further described in the paper [8]. The required accuracy is obtained in a frequent range by comparing the period of vibration in simulations to the experiment, illustrated in Fig 8. The experiment: Measure number 8: $m_Q = 0$ kg, ten vibrations are obtained in 22,62 s, followed by: $10 \cdot T_{EXP} = 176,77-154,15 = 22,62$ s, and the result is $T_{EXP} = 2,262$ s.

Figure 8 Experimental testing of the impulse regime: microdeformations of tower crane beam structure, measuring gauges MT1, MT2. Load mass $m_Q = 2200$ kg, orientation angle 65° radius, average period $T_0 = 2,262$ s

The period of vibration from the modal analysis is determined: ($m_Q = 0$ t), $T_{MOD} = 2,18726$ s, from Table 1. Relative digression of these two periods is given in equation (7) ε_T :

$$\varepsilon_T = \frac{T_{EXP} - T_{MOD}}{T_{EXP}} \cdot 100 = \frac{2,262 - 2,18726}{2,262} \cdot 100 = 3,304\% \quad (7)$$

where:

T_{EXP} - experimental period of vibration,

T_{MOD} - period of vibration from modal analysis.

The relative difference of vertical simulations of the end of the boom translations, experiment and simulation ($m_Q = 2200$ kg) is given in equation (8) ε_Z :

$$\varepsilon_T = \frac{Z_{EXP} - Z_{SIM}}{Z_{EXP}} \cdot 100 = \frac{0,68 - 0,6565}{0,680} \cdot 100 = +3,456\% \quad (8)$$

where:

Z_{EXP} - experimental value of vertical translation of the end of the boom,

Z_{SIM} - simulation value of vertical translation of the end of the boom.

After comparing the simulation diagram in Fig. 5 to the diagram of impulsive load lifting in Fig. 8, the similarity between the theoretical and experimental results is obvious. The diagram in Fig. 8 shows measured deformations during the lifting of the load in the core of the tower crane structure on measuring gauges MT1 and MT2. During the lifting of the load, the measured impulse was longer than during simulation (about 3 s). The results obtained by simulation correspond to the behavior of a real structure in terms of quality and size. Fig. 9 shows the photographs of the equipment and objects of experimental measurements.

Figure 7: Experimentally obtained deformations on tower structure MT1. Free tower crane vibration during the lowering of the load (Measure 8)

Figure 9 Photo graphics from the experimental testing of the tower crane on impulse regime and measuring equipment

8. CONCLUSION

The presented concept of simulation is the reality of modern and practical technology. This procedure can be successfully applied to the structures exposed to special actions, particularly incidental ones. Therefore, it can be concluded that:

The quality of simulation offers a possibility of complete reserving control, mainly due to thousands of elements in grid frame structures. Such a numeric extent could not be achieved by classic design.

Thus described transient model redefines the static model of structure design. The system of dynamic testing on special impacts is established, and it guarantees certainty in terms of aim and task.

Such a complex structure defines critical points of elements and critical effects. This information can provide monitoring. Location selection for monitoring should be performed on elements with the lowest level of reserving.

The observed structure of the tower crane is characterized by a high level of amplitude vibrations, which is a hardly acceptable category for human nature. The improvement is possible by increasing the geometric section of the tower, the boom and foundation.

The obtained structure displacement higher than 3% of geometry indicates that it is desirable to perform a non-linear transient analysis for better research quality and technical design, which is a new feature of modern software.

ACKNOWLEDGEMENTS

The paper is a part of the research performed within the project TR 35049, Faculty of Mechanical Engineering University of Niš. The authors would like to thank the Ministry of Education and Science of the Republic of Serbia.

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