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Primena hidrostatičkog klipnog uređaja za proveru sile pritezanja i dilatacije zavrtnja pri ostvarivanju zavrtanjske veze

Goran Mihajlović[1*], Milutin Živković[1], dr Milomir Gašić[2]
[1] Visoka tehnička mašinska škola strukovnih studija, Trstenik, Srbija
[2] Fakultet za mašinstvo i građevinarstvo u Kraljevu, Kraljevo, Srbija

Kod najodgovornijih zavrtanjskih veza (posebno kod onih sa velikim prednaprezanjem) veoma je važno da se tačno definiše potrebna sila pritezanja F_p koja ne sme biti manja od minimalne potrebne sile F_{pmin}, niti veća od maksimalne dozvoljene sile F_{pmax}. Ako bi bilo $F_p < F_{pmin}$ postojala bi realna opasnost od razdvajanja spojenih delova, a u suprotnom slučaju ako je $F_p > F_{max}$ moglo bi doći do plastičnih deformacija i time do trajnih oštećenja elemenata zavrtanjske veze.

Zato je potrebno silu pritezanja što preciznije definisati (propisati), a potom je u praksi na odgovarajući način meriti i kontrolisati pri ostvarivanju zavrtanjske veze.

U radu je izložena originalna laboratorijsko-eksperimentalna metoda za merenje sila i dilatacija zavrtnja prilikom pritezanja, zasnovana na primeni specijalnog hidrostatičkog klipnog uređaja, kao i kratak opis i princip rada uređaja.

Uređaj je realizovan na Visokoj tehničkoj mašinskoj školi strukovnih studija u Trsteniku i isti se može koristiti kako za laboratorijske vežbe u tehničkim školama i na fakultetima, tako i u industriji za proveru mehaničkih svojstava zavrtnjeva pre njihove ugradnje, kao i za razne eksperimente u istraživačkim laboratorijama i na institutima.

Ključne reči: zavrtanjska veza, sila pritezanja, dilatacija, hidrostatički uređaj, komparator, manometar, baždarenje.

0. UVOD

Zavrtanjske veze spadaju u red najčešće korišćenih mašinskih spojeva, zbog niza svojih specifičnosti i prednosti u odnosu na ostale spojeve: mogućnost tačnog relativnog pomeranja spojenih delova, lako pretvaranje obrtnog kretanja u translatorno ili obrnuto, čvrsto spajanje delova, pouzdano zaptivanje tj. hermetičnost, itd.

Međutim, s druge strane, zavrtanjska veza često puta može biti i najslabije mesto u celoj mašinskoj konstrukciji, pa se može desiti da sigurnost i pouzdanost u radu čitave konstrukcije zavise baš od sigurnosti i pouzdanosti zavrtanjske veze. Stoga zavrtanjske veze uvek predstavljaju *kritična mesta* u konstrukcijama kojima treba posvetiti posebnu pažnju pri projektovanju i konstruisanju mašinskih sistema [1, 2]. To znači da je potrebno najpre njihovo precizno geometrijsko dimenzionisanje i provera nosivosti, a potom i što preciznije definisanje i merenje sile pritezanja koja će obezbediti zahtevanu nosivost. Ovo je utoliko bitnije ukoliko je zavrtanjska veza odgovornija.

Da bi se bolje razumela osnovna funkcija i uloga zavrtanjske veze u okviru mašinskog sistema, potrebno je najpre istaći neke osnovne činjenice i specifičnosti navojnih spojeva u odnosu na ostale mašinske spojeve [3].

- Prema principu ostvarivanja spoja navojni spojevi predstavljaju posebnu vrstu kombinovanih *frikciono-prinudnih spojeva*, jer suštinu vezivanja spojenih delova predstavlja frikcija (trenje) između elemenata zavrtanjske veze, a takođe i specifičan geometrijski oblik (prinudni zahvat) spoljašnjeg i unutrašnjeg navoja.
- Prema mogućnosti razdvajanja delova nakon spajanja navojni spojevi su tipični *razdvojivi spojevi*, jer se elementi navojnog para mogu praktično neograničeni broj puta sklapati i rasklapati bez ikakvog oštećenja.
- Prema međusobnoj pokretljivosti spojenih delova u toku rada navojni spojevi mogu biti kako nepokretni, tako i pokretni. *Nepokretni (čvrsti) navojni spojevi* koriste se za spajanje delova u čvrstu celinu kojom se najčešće prenosi celokupno opterećenje između spojenih delova.

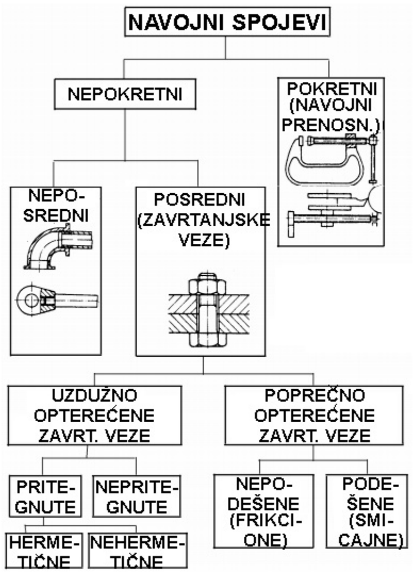
Kod pokretnih navojnih spojeva (navojnih prenosnika) prisutno je relativno kretanje tj. relativno pomeranje jednog elementa navojnog para u odnosu na drugi, što je u praksi našlo primenu kod: mernih instrumenata, ručnih alata, alatnih mašina i sličnih uređaja [4, 5, 6].

- Prema načinu ostvarivanja spoja-konstruktivnom izvođenju nepokretni navojni spojevi mogu se realizovati kao neposredni ili kao posredni spojevi. Kod *neposrednih navojnih spojeva* jedan element navojnog para snabdeven je spoljašnjim, a drugi element unutrašnjim navojem, tj. ovakav spoj nema nikakvih drugih međuelemenata. Kod *posrednih navojnih spojeva* ulogu posrednika imaju zavrtanj i navrtka, pa stoga ovi spojevi nose naziv *zavrtanjske veze*.
- Prema pravcu dejstva radnog opterećenja zavrtanjske veze dalje mogu biti *uzdužne* (sile dejstvuju u pravcu ose zavrtnja) i *poprečne* (sile dejstvuju poprečno u odnosu na osu zavrtnja). Prema zadatku koji obavljaju uzdužne veze mogu biti *pritegnute* (zahtevaju snažno pritezanje i iste se najčešće susreću u praksi) i *nepritegnute* (zbog svoje funkcije zahtevaju ili vrlo lagano ili nikakvo pritezanje). Poprečne zavrtanjske veze mogu biti *nepodešene* (zavrtanj se umeće u otvor sa zazorom jer je prečnik stabla zavrtnja manji od prečnika otvora) i *podešene* (nazivni prečnici zavrtnja i otvora su jednaki i propisuju se kao tolerisane mere, pa između njih nema nikakvog zazora već se njihove površine neposredno dodiruju).

*Kontakt adresa autora: Visoka tehnička mašinska škola strukovnih studija, Radoja Krstića 19, 37240 Trstenik, Srbija, goran.vtms@gmail.com

• Prema potrebi ostvarivanja zaptivenosti unutar spoja uzdužno opterećene zavrtanjske veze mogu biti *nehermetične-obične* (nema nekih posebnih zahteva u pogledu zaptivenosti spoja), ili *hermetične-pritisne* (potrebno je obezbediti dobru zaptivenost spoja da ne bi došlo do isticanja fluida iz nekog zatvorenog prostora).

Sve prethodno nabrojane vrste navojnih spojeva prikazane su na slici 1.



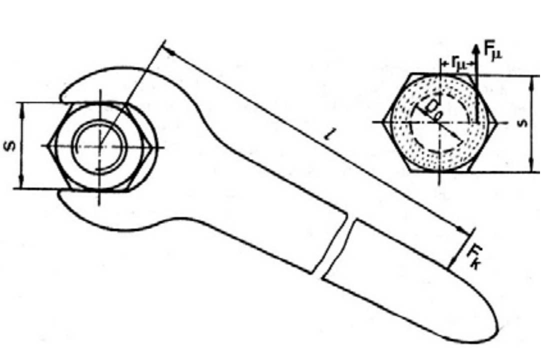
Sl. 1: *Podela navojnih spojeva*

1. OPŠTE POSTAVKE I NAČELA U VEZI PRITEZANJA ZAVRTANJSKIH VEZA

Zavrtanjska veza je rasklopivi navojni spoj koga čine zavrtanj, navrtka i spojeni delovi, a po potrebi tu još mogu biti i elementi za osiguranje i elementi za zaptivanje. Osnovni zadatak ove veze jeste da ista spreči bilo kakvo eventualno razdvajanje ili relativno pomeranje spojenih delova (ploča). Spomenuti zadatak ostvaruje se realizacijom dovoljno velike sile pritezanja F_p, čiji intenzitet zavisi kako od opštih uslova rada, tako i od odgovornosti funkcije koju obavlja zavrtanjska veza.

Sila pritezanja F_p generiše se dejstvom ključa na navrtku (ili glavu zavrtnja), što u njima prouzrokuje određeni moment pritezanja T_p koji je jednak proizvodu ručne sile na ključu F_k i kraka ključa l (slika 2):

$$T_p = F_k \cdot l \tag{1}$$



Sl. 2: *Ostvarivanje (pritezanje) zavrtanjske veze*

Jedan deo ostvarenog momenta pritezanja T_p utroši se na savladavanje otpora trenja između navojaka zavrtnja i navrtke (T_n). Drugi deo momenta pritezanja (T_μ) utroši se na savladavanje otpora trenja koji postoji na dodiru navtke i spojenih delova (ili glave zavrtnja i spojenih delova, u zavisnosti od toga šta se priteže). Na taj način jednačina raspodele ("utroška") ostvarenog momenta pritezanja ima sledeći oblik:

$$T_p = T_n + T_\mu \tag{2}$$

Iz teorije zavrtanjskih veza poznato je da spomenuti momenti iznose [7]:

$$T_n = F_p \cdot \frac{d_2}{2} \cdot tg(\varphi + \rho_n) \tag{3}$$

d_2-srednji-računski prečnik zavrtnja ($d_2 = \dfrac{d + d_3}{2}$);

d - veliki (nazivni) prečnik zavrtnja;

d_3 - mali prečnik (prečnik jezgra) zavrtnja;

φ - ugao uspona navoja ($tg\varphi = \dfrac{L}{d_2 \pi}$);

L - hod navoja ($L=z \cdot P$);

z - broj početaka navoja (uvek je z=1 kod zavrtanjskih veza, dok je $z \geq 1$ kod pokretnih spojeva);

P - korak navoja;

ρ_n - ugao trenja u navojnom paru zavrtanj-navrtka ($\rho_n = arctg\mu_n$);

μ_n - koeficijent trenja između navojaka zavrtnja i navrtke (zavisi od vrsta spregnutih materijala).

$$T_\mu = F_\mu \cdot r_\mu = F_p \cdot \mu \cdot r_\mu \tag{4}$$

F_μ - sila trenja na dodiru navrtke (ili glave zavrtnja) i spojenih delova (sl. 2);

μ - koeficijent trenja na dodiru navrtke (ili glave zavrtnja) i spojenih delova;

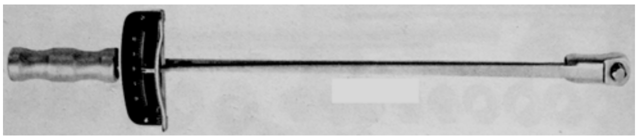
r_μ - srednji poluprečnik na kome dejstvuje rezultujuća sila trenja na dodiru navrtke (ili glave zavrtnja) i spojenih delova (sl. 2, $r_\mu = (s + D_0)/4$);

s - otvor ključa;

D_o - prečnik otvora u koji se ugrađuje zavrtanj.

Veličina sile na ključu F_k ograničena je snagom i jačinom ruke, dok krak ključa l može biti različit. U praksi postoji realna opasnost da radnik prilikom pritezanja proizvede nedovoljan moment pritezanja ako deluje suviše malom silom na ključ, ili da proizvede suviše veliki moment pritezanja ako deluje prevelikom silom na ključ. U prvom slučaju zavrtanjska veza biće *nedovoljno pritegnuta* čime se može ugroziti njena primarna radna funkcija, dok u drugom slučaju postoji opasnost da nastupi *plastična deformacija* ili čak i *lom zavrtnja*.

Zato je u cilju pravilne realizacije zavrtanjske veze potrebno da se moment pritezanja T_p ili sila pritezanja F_p na neki način mere i kontrolišu u toku pritezanja. Mnogo je češći slučaj da se meri moment pritezanja T_p uz pomoć specijalnih, tzv. *momentnih ključeva sa skalom* (slika 3), što je u industrijskoj praksi gotovo redovan slučaj.



Sl.3: *Momentni ključ sa skalom za merenje momenta pritezanja*

U ovom radu je, međutim, predložena jedna originalna laboratorijska eksperimentalna metoda za merenje sile pritezanja F_p, zasnovana na simultanoj primeni *hidrostatičkog klipnog uređaja* i *komparatora*. Njome se na veoma efikasan i relativno jednostavan način pre ugradnje u neku realnu konstrukciju može proveriti mehanički kvalitet (nosivost) zavrtnjeva, što je posebno značajno za najodgovornije zavrtanjske veze sa velikim prednaprezanjem.

2. ISO PREPORUKE ZA DEFINISANJE POTREBNE SILE PRITEZANJA ZAVRTANJSKIH VEZA

Potrebna sila pritezanja zavrtanjskih veza se u najopštijem slučaju može definisati na dva karakteristična načina i to [7]:

- Pomoću FAKTORA PRITEZANJA γ kojim se uzima u obzir pad sile pritezanja usled labavljenja i samoodvrtanja, što je samo približna i orijentaciona metoda;
- Pomoću STEPENA SIGURNOSTI PROTIV RAZDVAJANJA ZAVRTANJSKE VEZE S_r čije su ISO preporučene vrednosti date u tabeli T.1, što je znatno tačnija i preciznija metoda.

Tabela 1. *Preporučene vrednosti stepena sigurnosti S_r protiv razdvajanja zavrtanjske veze*

Odnos l_b/d	Dinamičko opterećenje											St. opt
	Red *"A"*		Red *"B"*			Red *"C"*			Red *"D"*			
	M4 M8	M10 M30	M4 M8	M10 M16	M18 M30	M4 M8	M10 M16	M18 M30	M4 M8	M10 M16	M18 M30	sve dim
Kratki <1	3	1,5	---	2	1,5	---	5	2	---	---	3,5	1,2
1...3	2	1,5	3	2	1,5	---	3	2	---	4	2,5	
Sred. 3...4	1,5	1,3	2	1,4	1,4	4	1,6	1,6	---	2	1,6	
4...6	1,5	1,3	2	1,4	1,4	2,5	1,6	1,6	---	2	1,6	
Dugi 6...8	1,4	1,3	1,6	1,3	1,3	2	1,4	1,4	4	1,6	1,6	
8...10	1,4	1,3	1,6	1,3	1,3	2	1,4	1,4	3	1,6	1,6	

Određivanje reda od *"A"* do *"D"*

Opterećenje zavrtnja	Klasa čvrstoće	do 8.8		10.9 do 12.9	
	Broj parova dodirnih površina	Hrapavost na dodiru spojenih delova			
		R_z=6,3μm	R_z=25μm	R_z=6,3μm	R_z=25μm
Uzdužno	do 3	B	C	A	B
	preko 3	C	D	B	C
Poprečno	do 3	C	D	B	C
	preko 3	D	---	C	D

♦ <u>Za obične (nehermetične) veze na koje deluje radna sila F_r, ali se ne zahteva garantovana sila F_b između spojenih delova</u>:

$$F_p \approx \gamma \cdot F_r\ , \qquad F_p = \xi_p \cdot S_r \cdot F_r \cdot (1 - \Phi) \qquad (5)$$

γ =1,5...2 ako je F_r statička, γ =2...4 ako je F_r dinamička sila;

ξ_p – faktor pritezanja zavistan od postupka montaže, a koji iznosi:

ξ_p =1 – za elektronsko merenje izduženja zavrtnja pri pritezanju;

ξ_p =1,6 – za merenje momenta pritezanja dinamometarskim ključem sa satnim mehanizmom;

ξ_p =2,5 – za pritezanje dinamometarskim ključem bez satnog mehanizma;

ξ_p =4 – za ručno pritezanje običnim ključem bez ikakvog merenja.

$\Phi' = c_z/(c_z + c_b)\ , \quad \Phi = q \cdot \Phi'$

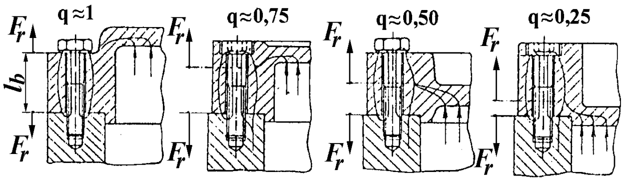
Φ' – faktor udela radne sile kojeg "prihvata" zavrtanj (teorijska–nekorigovana vrednost);

Φ – faktor udela radne sile kojeg "prihvata" zavrtanj (stvarna–korigovana vrednost);

c_z – krutost materijala zavrtnja;

c_b – krutost materijala spojenih delova (ploča);

q – faktor položaja napadne tačke radne sile (uvodi se u proračun zbog nepoklapanja napadnih tačaka sila F_p i F_r) – vrednosti ovog faktora mogu se približno odrediti prema slici 4.



Sl. 4: *Faktor položaja napadne tačke radne sile*

♦ <u>Za obične (nehermetične) veze na koje ne deluje nikakva radna sila F_r, ali se zbog ostvarivanja određene radne funkcije (na primer zadatog preklopa između spojenih delova) zahteva garantovana sila F_b između spojenih delova</u> (primer ovakve veze može biti konusni stezni spoj kojim se glavčina nekog obrtnog elementa montira na vratilo):

$$F_p \approx \gamma \cdot F_b\ ,\ F_p = \xi_p \cdot S_r \cdot F_b\ ,\ \text{где je}\ \gamma = 1{,}2...1{,}5. \quad (6)$$

♦ <u>Za pritisne (hermetične) veze na koje deluje radna sila F_r i od kojih se istovremeno zahteva garantovana radna sila F_b između spojenih delova</u>:

$$F_p \approx \gamma \cdot F_{bmin}\ , \quad F_p = \xi_p \cdot [F_b + S_r \cdot F_r \cdot (1 - \Phi)] \qquad (7)$$

$$F_{bmin} = p_{bmin} \cdot A'$$

A' – dodirna površina spojenih delova svedena po jednom zavrtnju;

p_{bmin} – minimalna vrednost površinskog pritiska na dodiru spojenih delova koja obezbeđuje zadovoljavajuće zaptivanje (zavisi od vrste materijala zaptivača);

* *Za nemetalne zaptivače*: p_{bmin} =(2...4)MPa za gumu, p_{bmin} =10MPa za teflon i slične materijale;

* *Za metalne zaptivače:* $p_{b\,\min} = m \cdot p_r \cdot \sqrt{b}$, где je: m=4 (za Al), m=5 (za Cu), m=6 (za meki čelik), m=7,5 (za tvrdi čelik); p_r – radni pritisak fluida u sudu pod pritiskom; b – širina zaptivača u cm.

* *Za neposredno zaptivanje čeličnih površina metal/na metal (bez zaptivača):* $p_{b\,\min} = m \cdot p_r \cdot \sqrt{b}$, где je: b – širina zaptivne površine u cm prema slici 5; faktor m se određuje u funkciji od radnog pritiska fluida u sudu pod pritiskom, prema tabeli T.2:

Tabela 2. *Vrednosti korekcionog faktora neposrednog zaptivanja metalnih površina m u funkciji od pritiska fluida*

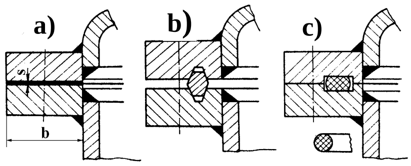
p_r [*bar*]	*< 40*	*40...80*	*80...120*	*120...200*	*200...350*	*350...600*	*600...800*
m=	*7*	*7...6*	*6...5*	*5...4*	*4...3*	*3...2*	*2...1,5*

Pored svih prethodno navedenih uslova, potrebno je da istovremeno bude ispunjen i sledeći uslov:

$F_p < F_{pmax}$ (8)

F_{pmax} – sila pritezanja koja bi dovela do trajnih plastičnih deformacija (gnječenja) površinskog sloja zaptivača (određuje se eksperimentalno za svaku konkretnu vrstu materijala zaptivača).

Osnovni principi zaptivanja kod hermetičnih zavrtanjskih veza prikazani su na sl. 5 (pljosnatim gumenim zaptivačem – slučaj ″a″; metalnim prstenom – slučaj ″b″; gumenim prstenom u posebnom žlebu – slučaj″ c″).



Sl. 5: *Osnovni principi zaptivanja kod hermetičnih zavrtanjskih veza*

♦ Poprečno opterećene nepodešene zavrtanjske veze (slika 6.a) zovu se još i *frikcione zavrtanjske veze*, jer se na dodiru spojenih delova javlja velika sila trenja F_μ koja se suprotstavlja poprečnoj radnoj sili $F_r = F_S$. Kod njih se zavrtanj umeće u otvor čiji je prečnik veći od prečnika stabla zavrtnja ($D_o > d$), a navrtka se mora pritegnuti snažno da bi se sprečilo proklizavanje spojenih delova usled dejstva radne sile. Potrebna sila pritezanja F_p se kod ovih veza sračunava po obrascu:

$F_p \geq (\xi_p \cdot S_\mu \cdot F_S) / (z \cdot i \cdot \mu)$ (9)

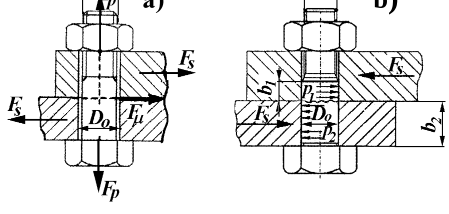
ξ_p – faktor pritezanja koji iznosi ξ_p =1,5...2 za statičku radnu silu ($F_S \approx$ const), odnosno ξ_p =2...4 za dinamičku radnu silu ($F_S \neq$ const);

S_μ =1,2...1,8 stepen sigurnosti protiv proklizavanja spojenih delova;

z – ukupan broj zavrtnjeva koji čine poprečno opterećenu nepodešenu zavrtanjsku vezu;

i – ukupan broj površina između spojenih delova na kojima se javlja sila trenja (otpor protiv proklizavanja);

μ – koeficijent trenja klizanja na dodiru spojenih delova (uzima se iz odgovarajućih tablica).



Sl. 6: *Poprečno opterećena nepodešena zavrtanjska veza (a)i poprečno opterećena podešena zavrtanjska veza (b)*

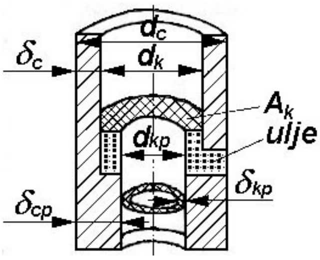
♦ Poprečno opterećene podešene zavrtanjske veze (slika 6.b) zovu se još i *smicajne zavrtanjske veze*, jer je usled dejstva poprečne radne sile $F_r = F_S$ stablo zavrtnja izloženo smicanju zbog njegovog neposrednog dodira sa površinom otvora u koji je zavrtanj umetnut ($D_o = d_s$, što znači da su ovde prečnici otvora i stabla zavrtnja tačno ″podešeni″ jedan prema drugome). Ovde se navrtka priteže lagano i to samo radi osiguranja spojenih delova od razmicanja, tako da normalni naponi u zavrtnju usled pritezanja ne prelaze 20% od granice tečenja dotičnog zavrtnja σ_{TM}:

$\sigma_p = (F_p / A_3) \leq 0{,}2 \cdot \sigma_{TM}$, $S_\sigma = (\sigma_{TM} / \sigma_p) \geq 5$ (10)

S_σ – parcijalni stepen sigurnosti zavrtnja s obzirom na normalne napone.

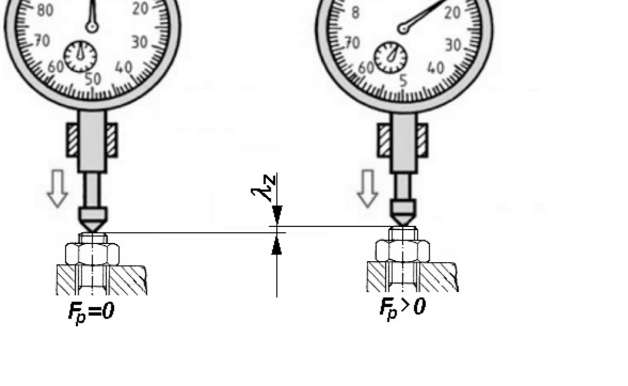
3. IZGLED, OPIS I PRINCIP RADA HIDROSTATIČKOG KLIPNOG UREĐAJA ZA MERENJE SILE I DILATACIJE ZAVRTNJA U TOKU PRITEZANJA

Stilizovani prikaz hidrostatičkog uređaja za merenje sila i dilatacija zavrtnja u toku pritezanja dat je na slici 7.

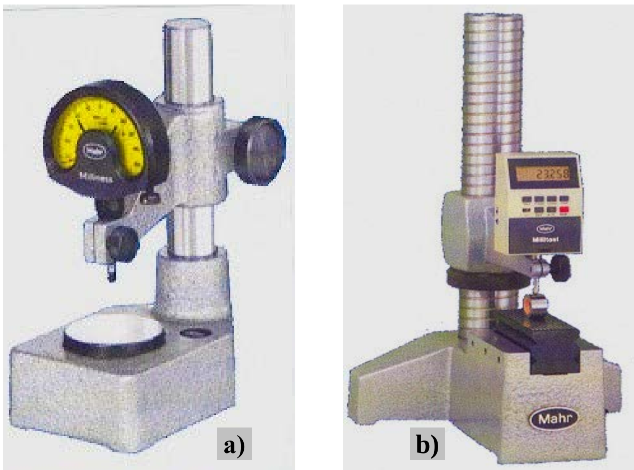


Sl. 7: *Stilizovani prikaz uređaja za merenje sile i dilatacije zavrtnja u toku pritezanja*

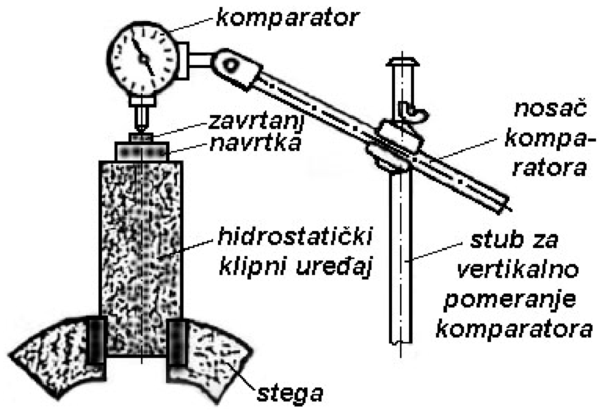
Princip rada hidrostatičkog uređaja za merenje sile i dilatacije zavrtnja u toku pritezanja zasniva se na činjenici da se usled pritezanja navrtke (ili glave zavrtnja) ključem zavrtanj elastično izdužuje za neku vrednost $+\lambda_z$, koja se može izmeriti pomoću komparatora (slika 8). U industrijskoj praksi najčešća je primena optičkih komparatora sa kružnom skalom i kazaljkom (slika 9.a), dok se za laboratorijska i naučna merenja i ispitivanja u raznim institutima sve više koriste elektronski komparatori sa digitalnim očitavanjem (slika 9.b). Šematski prikaz opreme za merenje sila i dilatacija zavrtnja pri pritezanju dat je na slici 10, na kojoj jedino nije prikazan manometar koji se prethodno mora izbaždariti u dekanjutnima (*daN*) radi direktnog očitavanja ostvarene sile pritezanja (F_p), [8, 9].



Sl. 8: *Merenje dilatacije zavrtnja pomoću komparatora*



Sl. 9:*Optički komparator (a) i elektronski komparator (b)*



Sl. 10: *Šematski prikaz opreme za merenje sile i dilatacije zavrtnja u toku pritezanja*

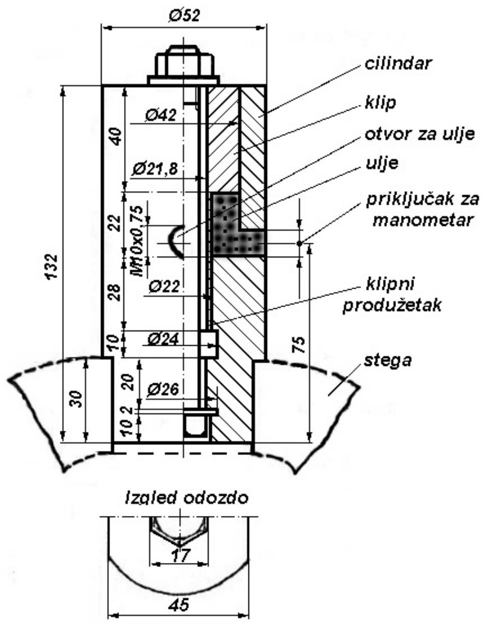
Glavni deo hidrostatičkog uređaja jeste cilindar čija debljina zida iznosi δ_c, u čijoj unutrašnjosti se nalazi šuplji klip sa klipnim produžetkom, pri čemu debljina zida cilindra na mestu gde naleže produžetak iznosi δ_{cp} (slika 7). Klip i cilindar ostvaruju neposredno naleganje metal/na metal (bez ikakvog zaptivača). S obzirom da u cilindru vlada vrlo visok pritisak, potrebno je radi što boljeg zaptivanja da njihove dodirne površine budu što finije obrađene i to najpre brušenjem, a potom i lepovanjem. Zato i viskozitet ulja treba da bude što veći da ulje ne bi prolazilo kroz zazor između klipa i cilindra, pa se zato preporučuje neko od ulja za amortizere, jer je ono najpodesnije za ove namene.

Kroz centralni otvor hidrostatičkog uređaja (kroz šupljinu klipa i klipnog produžetka) provlači se zavrtanj koji se ispituje (slika 11). U svom donjem delu otvor je šestougaonog oblika u koji se postavlja glava zavrtnja radi njenog pravilnog fiksiranja, odnosno radi sprečavanja njenog zaokretanja pri pritezanju. Pritezanjem navrtke prouzrokuje se pritisak na čelu klipa, koji će stoga težiti da sabije ulje u cilindru. Međutim, pošto je ulje praktično nestišljivo, u zavrtnju nastaje elastična deformacija u vidu njegovog izduženja $+\lambda_z$, a nastali pritisak ulja se shodno Paskalovom zakonu za tečnosti i nestišljive fluide prenosi na manometar, gde se direktno očitava kao ostvarena sila pritezanja u zavrtnju F_p. Pri tome treba strogo voditi računa da uređaj bude dobro fiksiran u stezi kako se ne bi pomerio pri pritezanju, jer bi to neminovno dovelo do pogrešnog očitavanja dilatacije zavrtnja na komparatoru.

Za dati eksperiment u principu se može upotrebiti zavrtanj bilo koje dužine i bilo kojeg (proizvoljnog) prečnika stabla. Treba imati u vidu da su zavrtnji manjeg prečnika i manje klase čvrstoće znatno ugroženiji od onih sa većim prečnikom i većom klasom čvrstoće [10]. U eksperimentu je odabran zavrtanj srednje klase čvrstoće sledeće oznake:

ZAVRTANJ M10x140-8.8 SRPS EN ISO 4016

Prilikom dimenzionisanja uređaja krenulo se od pretpostavke da će se nekada na njemu, pored onih najodgovornijih, takođe kontrolisati i zavrtnji sa znatno slabijim mehaničkih svojstvima, koji su u praksi i najugroženiji na lom. U njima se pritezanjem mogu generisati naponi pritezanja veći od granice kidanja, pa je se stoga krenulo od ideje da uređaj treba da bude u stanju da proizvede silu pritezanja koja će razoriti zavrtanj klase čvrstoće 5.6, čija minimalna zatezna čvrstoća iznosi $\sigma_M = R_m = 500 MPa$.



Sl. 11:*Mere i dimenzije hidrostatičkog uređaja za merenje sile i dilatacije zavrtnja u toku pritezanja*

Iz odgovarajućih standarda za metrički navoj M10 krupnog koraka pronalaze se sledeće vrednosti:

d=10mm-nazivni prečnik navoja;

d_2=9,026mm-srednji (računski) prečnik navoja;

d_3=8,16mm-prečnik jezgra navoja;

$A_3 = (d_3^2 \cdot \pi)/4 = 52{,}3 mm^2$-površina preseka jezgra zavrtnja;

P=1,5mm-korak navoja;

φ=3,028^o-ugao uspona navoja;

μ_n=0,15-koeficijent trenja klizanja u navojnom paru;

$\rho_n = arctg \mu_n = arctg 0{,}15 = 8{,}83^o$-ugao trenja u navojnom paru;

Maksimalna računska vrednost sile pritezanja koja bi pokidala zavrtanj M10 klase čvrstoće 5.6 (a koju treba da generiše hidrostatički uređaj) iznosi:

$$F_{pmax} = \sigma_M \cdot A_3 = 500 \cdot 52{,}3 = 26150 N = 2615 daN \quad (11)$$

Prečnici klipa (d_k) i klipnog produžetka (d_{kp}) usvajaju se empirijski, imajući u vidu činjenicu da se radi o zavrtnju nazivnog prečnika M10:

$$d_k = 42 mm \quad (12)$$

$$d_{kp} = 22 mm \quad (13)$$

4. POSTUPAK BAŽDARENJA MANOMETRA U CILJU MERENJA SILE PRITEZANJA

Potrebno je najpre izračunati površinu klipa A_k koja je dvostruko šrafirana na slici 7, a koja predstavlja površinu odgovarajućeg kružnog prstena:

$$A_k = \frac{d_k^2 \cdot \pi}{4} - \frac{d_{kp}^2 \cdot \pi}{4} = \frac{42^2 \cdot \pi}{4} - \frac{22^2 \cdot \pi}{4} \approx 1000mm^2 \quad (14)$$

Maksimalni pritisak u cilindru pri kidanju zavrtnja M10-5.6 bi iznosio:

$$p_{\max} = \frac{F_{p\max}}{A_k} = \frac{26150}{1000} \approx 26MPa \approx 260bar \quad (15)$$

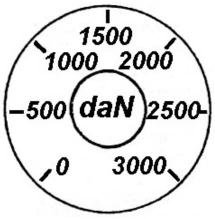
Na osnovu (15) može se zaključiti da je potrebno odabrati manometar koji može da meri pritiske do 30*MPa*, odnosno do 300*bar*.

Baždarenje manometra treba izvršiti tako što će se na skali umesto svake postojeće vrednosti za pritisak napisati korespodentna vrednost sile pritezanja F_p, shodno vrednostima navedenim u tabeli T.3.

Tabela. 3. *Korespodentne vrednosti pritiska i sile pritezanja potrebne za baždarenje manometra*

Postojeća vrednost pritiska p [MPa] / ([bar])	Površina klipa A_k [mm^2]	Sila pritezanja F_p [N]	Sila pritezanja F_p [daN]
5 / (50)	1000	5000	500
10 / (100)	1000	10000	1000
15 / (150)	1000	15000	1500
20 / (200)	1000	20000	2000
25 / (250)	1000	25000	2500
30 / (300)	1000	30000	3000

To praktično znači da će se na manometru umesto očitavanja pritiska od 50*bar* očitavati sila pritezanja od 500*daN*, umesto očitavanja pritiska od 100*bar* sila pritezanja od 1000*daN*, ... , umesto očitavanja pritiska od 300*bar* sila pritezanja od 3000*daN* (slika 12).



Sl. 12: *Manometar za merenje sile pritezanja zavrtnja*

Sada kada je poznata maksimalna vrednost pritiska koju može generisati uređaj, mogu se konačno odrediti i debljina zida šupljeg klipnog produžetka δ_{kp}, kao i debljine zida cilindra δ_c i δ_{cp} prikazane na slici 7:

$$\delta_{kp} = \frac{d_{kp} \cdot p_{\max}}{2 \cdot \sigma_{zdoz}} = \frac{22 \cdot 30}{2 \cdot 210} = 1{,}57mm$$

$$\delta_c = \frac{d_k \cdot p_{\max}}{2 \cdot \sigma_{zdoz}} = \frac{42 \cdot 30}{2 \cdot 210} = 3mm$$

$\sigma_{zdoz}=210MPa$-dozvoljena vrednost napona na zatezanje konstrukcijskog čelika Č0745 (materijala klipa i cilindra).

S obzirom na preporučenu vrednost stepena sigurnosti za debljinu zida S=1,5...2 (u pitanju je statičko naprezanje), konačno se mogu usvojiti sledeće debljine zidova:

$\delta_{kp}=3mm$ (16)

$\delta_c=5mm$ (17)

Spoljašnji prečnik cilindra (d_c):

$d_c=d_k+2\delta_c=42+2\cdot 5=52mm$ (18)

$$\delta_{cp} = \frac{d_c - d_{kp}}{2} = \frac{52-22}{2} = 15mm \quad (19)$$

5. TEORIJSKA VREDNOST SILE PRITEZANJA I DILATACIJE ZAVRTNJA

Osnovni cilj izvedenih eksperimenata bio je taj da se izvrše sledeće dve provere:

- da li teorijski sračunatoj vrednosti sile pritezanja F_{pt} odgovara stvarna vrednost sile pritezanja F_{ps} očitana na manometru hidrostatičkog uređaja;
- da li teorijski sračunatoj vrednosti dilatacije zavrtnja λ_{zt} odgovara stvarna vrednost dilatacije zavrtnja λ_{zs} izmerena pomoću komparatora.

Teorijska vrednost sile pritezanja dobija se na osnovu jednačine (2), imajući u vidu da je drugi član na desnoj strani jednačine jednak nuli (T_μ=0). Razlog je taj što se sila trenja u toku pritezanja ne može preneti sa navrtke na čelo klipa uređaja, zato što se klip zajedno sa navrtkom slobodno obrće (rotira) oko svoje vertikalne ose. Na taj način će moment pritezanja biti jednak momentu potrebnom za savladavanje trenja između navojaka zavrtnja i navrtke:

$$T_p = T_n = F_{pt} \cdot \frac{d_2}{2} \cdot tg(\varphi + \rho_n) = F_{pt} \cdot \frac{0{,}009026}{2} \cdot tg(3{,}028 + 8{,}53)$$

$$T_p = 0{,}000923 \cdot F_{pt} \quad [daNm] \quad (20)$$

Za odgovarajuću zadatu vrednost momenta pritezanja T_p može se sračunati teorijska vrednost sile pritezanja tavrtnja pomoću sledećeg obrasca:

$$F_{pt} = \frac{T_p}{0{,}000923} = 1083{,}5 \cdot T_p \quad [daN] \quad (21)$$

Teorijska vrednost dilatacije zavrtnja λ_{zt} računa se na osnovu Hukovog zakona, prema kome je napon u zavrtnju usled pritezanja σ_{zp} srazmeran dilataciji zavrtnja koja se može izraziti i preko relativnog izduženja ε_z:

$$\sigma_{zp} = E_z \cdot \varepsilon_z = E_z \cdot \frac{\lambda_{zt}}{l} = 2{,}1 \cdot 10^5 \cdot \frac{\lambda_{zt}}{120} = 1750 \cdot \lambda_{zt} \quad [MPa]$$

$E_z=2{,}1\cdot 10^5 MPa$-modul elastičnosti materijala zavrtnja (čelik);
$l\approx 120mm$-dužina zavrtnja između glave zavrtnja i navrtke (debljina spojenih delova, prema slici 11).

$$\lambda_{zt} = \frac{\sigma_{zp}}{1750} = \frac{F_{pt}}{A_3 \cdot 1750} = \frac{F_{pt}}{52{,}3 \cdot 1750} = 0{,}000011 \cdot F_{pt}$$

gde je F_{pt} izraženo u njutnima [N].

Pošto je manometar na hidrostatičkom uređaju baždaren u dekanjutnima [daN], važiće sledeća relacija:

$$\lambda_{zt} = 0{,}00011 \cdot F_{pt} = 1{,}1 \cdot 10^{-4} \cdot F_{pt} \quad [mm] \quad (22)$$

6. TABELARNI I GRAFIČKI PRIKAZ IZMERENIH REZULTATA

Prilikom pritezanja navrtke (ili glave zavrtnja) opterećenja se prenose putem dodira i to parcijalno sa navojaka navrke (ili sa navojaka glave zavrtnja) na navojke spregnutog elementa navojnog para. Najveća su na dodiru prvih aktivnih navojaka najbližih pločama (u našem primeru najbližih klipu hidrostatičkog cilindra), a postepeno se smanjuju prema gornjim navojcima, koji su na taj način znatno rasterećeniji u odnosu na donje navojke.

Eksperiment se izvodi tako što se postepeno povećava intenzitet sile pritezanja F_p i to u nekoliko odabranih "koraka", pri čemu je naš predlog da to bude u šest "koraka" jer na manometru postoji isto toliko izbaždarenih vrednosti (slika 12).

Sračunate teorijske i izmerene stvarne vrednosti sile pritezanja prikazane su u tabeli T.4, a sračunate teorijske i izmerene stvarne vrednosti dilatacije zavrtnja prikazane su u tabeli T.5.

Tabela 4. *Teorijske (računske) i stvarne (izmerene) vrednosti sile pritezanja zavrtnja*

Redni broj merenja	Moment pritezanja* $T_p\,[daNm]$	Sila pritezanja** $F_{pt}\,[daN]$	Sila pritezanja*** $F_{ps}\,[daN]$	Greška $\lvert F_{pt} - F_{ps}\rvert$ $[daN]$	Greška $\lvert F_{pt} - F_{ps}\rvert$ $[\%]$
1	0,46	500	560	60	12
2	0,92	1000	880	120	13,6
3	1,38	1500	1450	50	3,4
4	1,85	2000	2100	100	5
5	2,31	2500	2350	150	6,4
6	2,77	3000	2750	250	9,1

* Na osnovu formule (20) pri teorijskom sračunavanju, a pri merenju očitava se na skali momentnog ključa;

** Na osnovu formule (21);

*** Očitava se na manometru hidrostatičkog uređaja.

Tabela 5. *Teorijske (računske) i stvarne (izmerene) vrednosti dilatacije zavrtnja u toku pritezanja*

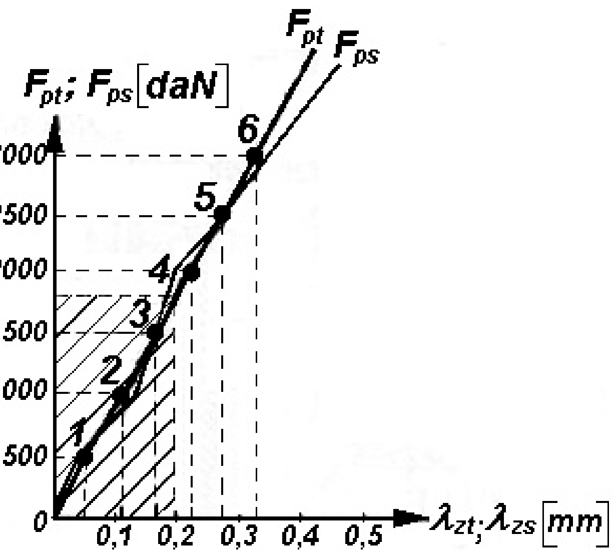
Redni broj merenja	Sila pritezanja* $F_p\,[daN]$	Dilatacija zavrtnja** $\lambda_{zt}\,[mm]$	Dilatacija zavrtnja*** $\lambda_{zs}\,[mm]$	Greška $\lvert\lambda_{zt} - \lambda_{zs}\rvert$ $[mm]$	Greška $\lvert\lambda_{zt} - \lambda_{zs}\rvert$ $[\%]$
1	500	0,055	0,040	0,015	37,5
2	1000	0,110	0,130	0,020	18,2
3	1500	0,165	0,180	0,015	9,1
4	2000	0,220	0,190	0,030	15,8
5	2500	0,275	0,290	0,015	5,4
6	3000	0,330	0,360	0,030	9,1

* Pri merenju očitava se na manometru hidrostatičkog uređaja;

** Na osnovu formule (22);

*** Očitava se na komparatoru.

Dobijeni rezultati iz tabela T.4 i T.5 mogu se grafički prikazati pomoću *deformacionog dijagrama zavrtanjske veze* (slika 13). Na ordinatnu osu ovog dijagrama nanose se vrednosti sile pritezanja u *daN*, a na apscisu dilatacije zavrtnja u *mm*. Spajanjem tačaka sa koordinatama [λ_{zt} ; F_{pt}] za svako konkretno merenje dobija se prava linija koja prikazuje *teorijsku deformaciju zavrtnja*. Analognim spajanjem tačaka sa koordinatama [λ_{zs}; F_{ps}] za svako konkretno merenje dobija se takođe prava linija koja u tom slučaju prikazuje *stvarni tok deformacije zavrtnja* u toku njegovog pritezanja.



Sl. 13: *Deformacioni dijagram zavrtnja M10 klase čvrstoće 8.8 u toku pritezanja*

7. ZAKLJUČAK

Prethodno opisana metoda merenja sile pritezanja i dilatacije zavrtnja razvijena je na Visokoj tehničkoj mašinskoj školi strukovnih studija u Trsteniku. Ista se može aplicirati u praksi kao originalna laboratorijsko-eksperimentalna metoda. Njome se na relativno jednostavan i brz način pomoću hidrostatičkog uređaja prikazanog na slici 11 mogu proveravati zavrtnji raznih dimenzija i različitih klasa čvrstoće, kod svih vrsta prednapregnutih zavrtanjskih veza. Ovakve provere u praksi mogu imati višestruki značaj [11, 12]:

• Da učenicima i studentima na tehničkim školama i na fakultetima [13] na očigledan način demonstriraju značaj pravilnog odabiranja sile pritezanja, koja se uvek mora nalaziti u zadatom opsegu [14, 15]:

$$F_{pmin} \le F_p \le F_{pmax} \tag{23}$$

Ukoliko je zavrtanjska veza nedovoljno pritegnuta ($F_p < F_{pmin}$) moguće je da u toku radnog veka dođe do razdvajanje spojenih delova. U suprotnom slučaju, ako je ista suviše pritegnuta ($F_p > F_{pmax}$) može doći do trajnih, tj. plastičnih deformacija bilo zavrtnja, bilo ostalih elemenata zavrtanjske veze. Zato se stvarna vrednost sile pritezanja sme nalaziti samo u *elastičnom području*, a to znači samo u šrafiranoj oblasti na deformacionom dijagramu zavrtnja prikazanom na slici 13.

• U industriji se pomoću navedene metode mogu upoređivati zavrtnji istih dimenzija i iste klase čvrstoće, ali od različitih proizvođača. Za ovo je najpre potrebno izvršiti pravilno *uzorkovanje*, tj. odabiranje za analizu optimalnog broja zavrtnjeva od svakog proizvođača ponaosob, na osnovu veličine serije kupljenih zavrtnjeva. Time se mogu brzo identifikovati proizvođači čiji su zavrtnji značajno nižeg kvaliteta od onog zahtevanog standardima, kao i proizvođači čiji će kvalitet možda biti i veći od uobičajenog (prosečnog) kvaliteta na tržištu [16, 17, 18, 19].

• U istraživačkim laboratorijama i na institutima privrednih preduzeća moguće je osmisliti i sasvim nove i *originalne eksperimente*, koji imaju za cilj dublje istraživanje neke specifične problematike u vezi rada i ostvarivanja zavrtanjskih veza. Tako se na primer mogu istraživati: uticaj povišenih (ili sniženih) temperatura na dilatacije zavrtnja prilikom pritezanja; uticaj koraka navoja (fini, grubi) na sile i dilatacije zavrtnja; uticaj dužine nošenja navojnog spoja (broja navojaka zavrtnja i navrtke u aktivnom dodiru) na sile i dilatacije u zavrtnju; uticaj vibracija u toku pritezanja na sile i deformacije u zavrtnju, itd. [20, 21, 22].

Na kraju treba istaći činjenicu da je osim zavrtnja *M10* moguće pomoću navedene metode ispitivati i zavrtnje većih ili manjih *nazivnih prečnika* i većih ili manjih *dužina* od onih navedenih u eksperimentu. Jedina je razlika ta što bi u tom slučaju mere i dimenzije hidrostatičkog uređaja prikazane na slici 11 trebalo prilagoditi prečniku i dužini zavrtnja koji se ispituje, dok bi princip merenja sila i dilatacija, a takođe i baždarenje manometra, ostali nepromenjeni.

8. LITERATURA

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Application of the Hydrostatic Piston Device for Testing the Force of Tightening and Screw Dilatation in a Screw Connection

Goran Mihajlović[1*], Milutin Živković[1], Milomir Gašić[2]
[1] Higher Technical School of Mechanical Engineering, Trstenik (Serbia)
[2] Faculty of mechanical and Civil Engineering in Kraljevo, Kraljevo (Serbia)

For the most responsible screw connections (especially those with large prestressing) it is very important to precisely define the required tightening force F_p which must not be less than the minimum necessary force F_{pmin}, nor greater than the maximum permissible force F_{pmax}. If $F_p < F_{pmin}$ then there would be a real danger of shifting between connected parts, while in the opposite case if $F_p > F_{max}$ plastic deformations may occur and consequently permanent damage of some elements of screw connection.Therefore, the tightening force should be defined as precisely as possible, and then in practice it needs to be adequately measured and controlled in a screw connection. The paper presents an original laboratory-experimental method for measuring the forces and dilatations of the screw during tightening based on the application of a special hydrostatic piston device as well as a brief description and operating principle of the device. The device was designed at the Higher Technical School of Mechanical Engineering in Trstenik and it can be used for laboratory exercises in technical schools and faculties, as well as in the industry for checking the mechanical properties of the screws before being mounted, and also for various experiments in research laboratories and institutes.

Key words: screw connection, force of tightening, dilatation, hydrostatic device, comparator, manometer, calibration.

0. INTRODUCTION

Screw connection is one of most commonly used mechanical joints, due to a number of specific features and advantages over other joints, e.g. the possibility of accurate relative displacement of connected parts, easy conversion of rotary motion into translational motion or vice versa, tight joining of the parts, reliable sealing, i.e. tightness, etc.

On the other hand, screw connection can often be the weakest place in the entire machine construction, so it is possible that during work the safety and reliability of the entire construction depends almost exclusively on the safety and reliability of the screw connection. Therefore, screw connections always represent critical points in structures and thus draw attention during the design and construction of the machine systems [1, 2]. This means that their precise geometric dimensioning and load capacity verification are necessary, and then a precise definition and measurement of the tightening force that will provide the required load capacity. This is more important if the connection is more responsible.

To better understand the basic function and role of the screw connection within the machine system, it is necessary first to point out some basic facts and specificities of threaded joints in relation to other machine joints [3].

- According to the principle of jointing, the threaded joints represent a special type of combined friction-compulsive joints, because the essence of joining connected parts is the friction between the connecting elements, and also a specific geometric shape (compulsive contact) of the external and internal thread.
- According to the possibility of splitting the parts after joining, the threaded joints are typical separable joints, because the elements of threaded pairs can be assembled and disassembled practically in an unlimited number of times without any damage.
- According to the relative mobility of connected parts during operation, the threaded joints may be either motionless or motional. Motionless (strong) thread joints are used for joining parts into a solid part that often transfers the entire load between the joined parts. With motional thread joints (thread transmissions) there is a relative movement, i.e. a relative displacement of one element of the pair compared to the other, which in practice has been applied to measuring instruments, hand tools, machine tools and similar devices [4, 5, 6].
- According to the manner of constructive design, strong threaded joints can be direct or indirect. When threaded joint is direct, one threaded element is supplied with an outer thread and the second element is supplied with the internal thread, i.e. this joint has no other additional element. When threaded joint is indirect, screw and screw-nut have mediating function, so these joints are called the screw connections.
- Depending on the direction of work forces, the screw connections can be longitudinal (forces acting in the direction of screw) and transverse (forces acting transversely in relation to the axis of screw). According to the operating function longitudinal screw connections may be tightened (they require strong tightening and they are most commonly encountered in practice) and nontightened (because of their function they require very light or no tightening). Transverse screw connections can be nonadjusted (screw is inserted into the hole with gap because the screw stem diameter is smaller than the diameter of holes) and adjusted (the nominal diameters of the screw

*Corresponding author:Higher Technical School of Mechanical Engineering, Radoja Krstića 19, 37240 Trstenik, Serbia,goran.vtms@gmail.com;

and the hole are equal and they are prescribed as tolerated dimensions, so there is no gap between them but their surfaces directly touch each other).

- According to the criterion of achieving sealing within the joint, longitudinal screw connections can be nonhermetic-ordinary (there are no special requirements related to sealing), or hermetic-compressive (it is necessary to ensure a good tightness of the joint to avoid fluid leakage from a closed area).

All the previously mentioned types of threaded joints are shown in Fig. 1.

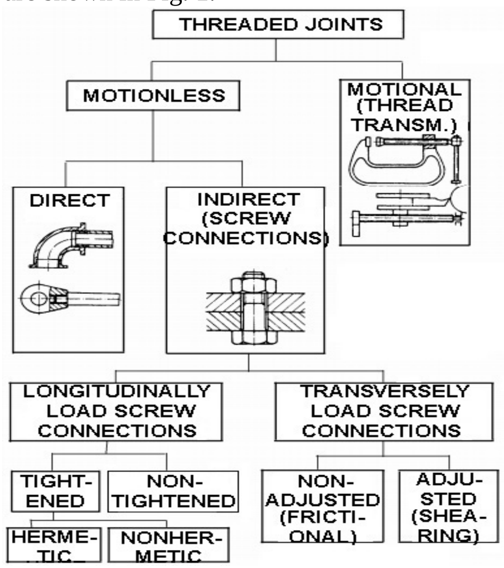

Fig. 1: *Classification of threaded joint*

1. GENERAL ITEMS AND PRINCIPLES REFERRING TO TIGHTENING OF SCREW CONNECTION

The screw connection is demountable threaded joint which consists of screw, screw-nut and connected parts, and there are insurance elements and sealing elements, if necessary. The basic task of this connection is to prevent any possible separation or relative displacement of connected parts (boards). This task is accomplished by a large enough tightening force F_p, whose intensity depends both on the general working conditions and the responsibility of the function performed by the screw connection.

Tightening force F_p is generated by the action of the wrench on the nut-screw (or on screw head), which causes a certain tightening torque T_p equal to the product of manual force on the wrench F_k and wrench length l (Fig. 2):

$$T_p = F_k \cdot l \tag{1}$$

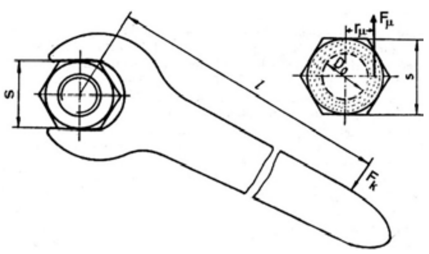

Fig. 2: *Tightening of a screw connection*

One part of the tightening torque T_p is spent to overcome the resistance of friction inside the threads between screw and screw-nut (T_n). The second part of the resulting tightening torque (T_μ) is spent to overcome the resistance of friction which occurs at the place of contact between screw-nut and connected parts (or between screw head and connected parts, depending on what is being tightened). In that way, equation of distribution of resulting tightening torque has the following form:

$$T_p = T_n + T_\mu \tag{2}$$

From the theory of screw connections, it is known that the mentioned moments have the following form [7]:

$$T_n = F_p \cdot \frac{d_2}{2} \cdot tg(\varphi + \rho_n) \tag{3}$$

d_2- medium-computational diameter of the screw ($d_2 = \frac{d + d_3}{2}$);

d - large (nominal) screw diameter;

d_3 – small diameter (core diameter) of the screw;

φ - angle of the inclination of thread ($tg\varphi = \frac{L}{d_2 \pi}$);

L – pace of the thread ($L = z \cdot P$);

z – number of thread beginnings (it is always $z=1$ for screw connections, while it is $z \geq 1$ for motional threaded joints);

P – pitch of the thread;

ρ_n - angle of friction in the threaded pair screw-screw-nut ($\rho_n = arctg\mu_n$);

μ_n - coefficient of friction between the screw threads and screw-nut threads (it depends on the types of bonded materials).

$$T_\mu = F_\mu \cdot r_\mu = F_p \cdot \mu \cdot r_\mu \tag{4}$$

F_μ - friction force at the place of contact of nut-screw (or screw head) and connected parts (Fig. 2);

μ - friction coefficient at the place of contact of nut-screw (or screw head) and connected parts;

r_μ - the middle radius on which the resulting force of friction acts at the place of contact of nut-screw (or screw head) and connected parts (Fig. 2, $r_\mu = (s + D_0)/4$);

s – wrench hole;

D_o - diameter of the hole in which the screw is mounted.

The size of the force on the wrench F_k is limited by power and strength of the arm, while the wrench length l can be different. In practice there is a real danger that, during tightening, the worker produces an insufficient tightening torque if he produces too little force on wrench, or to produce too high tightening torque if he causes too much force on wrench. In the first case the screw connection will be insufficiently tightened which may compromise its primary operating function, while in the second case there is a risk of plastic deformation or even breakage of the screw.

Therefore, in order to properly make the screw connection, it is necessary that tightening torque T_p or tightening force F_p, are somehow measured and controlled during tightening. It is more often to measure the tightening torque T_p by means of special torque wrench with scale (Fig. 3), which is almost a usual case in practice.

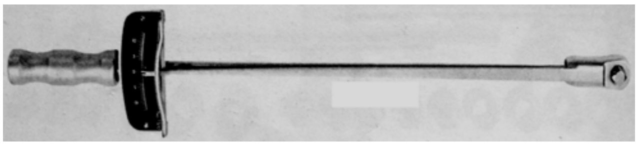

Fig. 3: *Torque wrench with scale for measuring the torque of tightening*

In this paper, however, one original laboratory experimental method for measuring the tightening force F_p is proposed, and it is based on simultaneous application of hydrostatic piston device and comparator. Using this method it is possible to efficiently and simply check the mechanical quality (load capacity) of the screws before they are mounted into a real construction, which is especially important for the most responsible screw connections with high preloading.

2. ISO RECOMMENDATIONS FOR DEFINING THE NECESSARY FORCE OF TIGHTENING OF A SCREW CONNECTION

The necessary force of tightening the screw connections in the most general case can be defined in two characteristic ways [7]:

- using *tightening factor* γ which takes into account the decrease of the force of tightening due to loosening and self-unscrewing, which is just an approximate and orientation method;
- using *safety level* S_r *in order to prevent separation of the screw connection* whose ISO recommended values are given in Table 1, which is a more accurate and more precise method.

Table 1. *Recommended safety level values* S_r *in order to prevent* separation *of the screw connection*

Quotient l_b/d	Dynamic load											St. load
	Ser. *"A"*		Series *"B"*			Series *"C"*			Series *"D"*			
	M4 M8	M10 M30	M4 M8	M10 M16	M18 M30	M4 M8	M10 M16	M18 M30	M4 M8	M10 M16	M18 M30	all dim
Short <1	3	1,5	---	2	1,5	---	5	2	---	---	3,5	
1...3	2	1,5	3	2	1,5	---	3	2	---	4	2,5	
Mid. 3...4	1,5	1,3	2	1,4	1,4	4	1,6	1,6	---	2	1,6	1,2
4...6	1,5	1,3	2	1,4	1,4	2,5	1,6	1,6	---	2	1,6	
Long 6...8	1,4	1,3	1,6	1,3	1,3	2	1,4	1,4	4	1,6	1,6	
8...10	1,4	1,3	1,6	1,3	1,3	2	1,4	1,4	3	1,6	1,6	

Determining the order from *"A"* to *"D"*

Screw load	Strength class	to 8.8		10.9 to 12.9	
	Number of pairs of contact surfaces	Roughness on the touch of connected parts			
		R_z=6,3μm	R_z=25μm	R_z=6,3μm	R_z=25μm
Longitudinally	up to 3	B	C	A	B
	more than 3	C	D	B	C
Transversely	up to 3	C	D	B	C
	more than 3	D	---	C	D

♦ For ordinary (nonhermetic) connections on which the work force F_r is acting, but the guaranteed force F_b between the fused parts is not required:

$$F_p \approx \gamma \cdot F_r \ , \qquad F_p = \xi_p \cdot S_r \cdot F_r \cdot (1 - \Phi) \qquad (5)$$

γ =1,5...2 if F_r is static force, γ =2...4 if F_r is dynamic force;

ξ_p – tightening factor that depends on the assembly process, which has the following values:

ξ_p =1 – for precise electronic measurement of the elongation of the screw during tightening;

ξ_p =1,6 – for measuring the torque of tightening with a dynamometer wrench with clock mechanism;

ξ_p =2,5 – for tightening with a dynamometer wrench without clock mechanism;

ξ_p =4 – for manual tightening with ordinary wrench without any measurement.

$$\Phi' = c_z /(c_z + c_b) \ , \quad \Phi = q \cdot \Phi'$$

Φ' – factor of share of working force which the screw accepts (theoretical-uncorrected value);

Φ – factor of share of working force which the screw accepts (real-corrected value);

c_z – stiffness of screw material;

c_b – stiffness of the materials of the connected parts (boards);

q – factor of the position of the point of application of the workforce (it is introduced into the calculation due to the incompatibility of the points of application of the forces F_p and F_r) – the values of this factor can be approximately calculated according to Fig. 4.

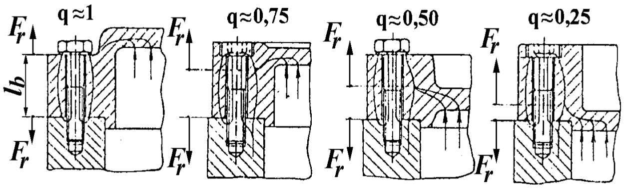

Fig. 4:*Position factor of point of application of the workforce*

♦ For ordinary (nonhermetic) connections on which the work force F_r is not acting, but due to the achievement of a certain working function (for example, the prescribed lap between connected parts) a guaranteed force F_b between connected parts is required (an example of this connection can be the conical clamping joint by which the hub of a rotary element is mounted on the shaft):

$$F_p \approx \gamma \cdot F_b \ , \ F_p = \xi_p \cdot S_r \cdot F_b, \text{ where } \gamma = 1{,}2...1{,}5. \qquad (6)$$

♦ For compressive (hermetic) connections on which the work force F_r is acting and simultaneously guaranteed force F_b between connected parts is required:

$$F_p \approx \gamma \cdot F_{bmin} \ , \quad F_p = \xi_p \cdot [F_b + S_r \cdot F_r \cdot (1 - \Phi)] \qquad (7)$$

$$F_{bmin} = p_{bmin} \cdot A'$$

A' – the contact surface of the fused parts reduced per one screw;

p_{bmin} – minimum value of surface pressure on the contact of connected parts which ensures satisfactory sealing (depends on the type of sealing material);

* *For non-metallic seals*: p_{bmin} =(2...4)MPa for rubber, p_{bmin} =10MPa for teflon and similar materials;

* *For metallic seals:* $p_{b\min} = m \cdot p_r \cdot \sqrt{b}$, where: m=4 (for Al), m=5 (for Cu), m=6 (for soft steel), m=7,5 (for hard steel); p_r – operating pressure of the fluid in the pressurized vessel; b – width of the seal in cm.

* *For direct sealing of steel surfaces metal/on metal (without seals):* $p_{b\,\min} = m \cdot p_r \cdot \sqrt{b}$, where: b – width of the sealing surface in *cm* according to Figure 5; factor *m* is determined in function of the working pressure of the fluid in the pressurized vessel, according to Table 2:

Table 2. *The values of correction factor of immediate sealing of metal surfaces m in function of fluid pressure*

p_r [bar]	< 40	40...80	80...120	120...200	200...350	350...600	600...800
m=	7	7...6	6...5	5...4	4...3	3...2	2...1,5

In addition to all of the above conditions, the following condition is required at the same time:

$$F_p < F_{pmax} \qquad (8)$$

F_{pmax} – tightening force that would cause permanent plastic deformations of the sealing surface of the sealant (it is determined experimentally for each specific type of sealing material).

The basic sealing principles for hermetic screw connections are shown in Fig. 5 (flat rubber seal – the case ″a″; metal ring – the case ″b″; a rubber ring in a special groove – the case ″c″).

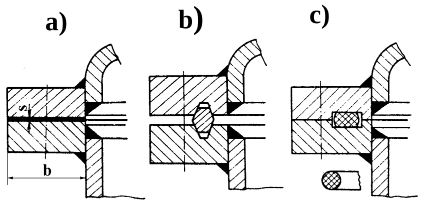

Fig. 5: *Basic sealing principles which are used for hermetic screw connections*

♦ <u>Transversely loaded nonadjusted screw connections</u> (Fig. 6.a) are also called *friction screw connections*, because at the touch of the connected parts there is a high friction force F_μ which opposes the transversal workforce $F_r = F_S$. In them, the screw is inserted into an opening whose diameter is greater than the diameter of screw body ($D_o > d$), and the screw-nut must be tightened tightly to prevent the slipping of parts due to the effect of the work force. The required force of tightening F_p for these connections is calculated according to the formula:

$$F_p \geq (\xi_p \cdot S_\mu \cdot F_S) / (z \cdot i \cdot \mu) \qquad (9)$$

ξ_p – tightening factor which has the values:

ξ_p =1,5...2 for static working force ($F_S \approx$ const),

ξ_p =2...4 for dynamic work force ($F_S \neq$ const);

S_μ =1,2...1,8 – degree of security against slipping of connected part;

z – the total number of screws which make transversely loaded nonadjusted screw connection;

i – the total number of surfaces between the joined parts where the friction force occurs (resistance against slipping);

μ – friction coefficient of slipping on the touch of connected parts (it is defined in the appropriate tables).

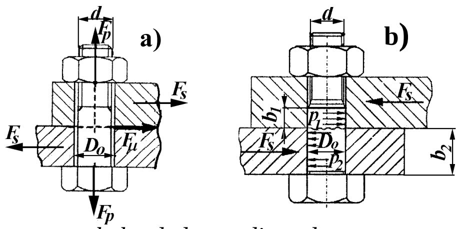

Fig. 6: *Transversely loaded nonadjusted screw connection (a) and transversely loaded adjusted screw connection (b)*

♦ <u>Transversely loaded adjusted screw connections</u> (Fig. 6.b) are also called *shearing screw connections*, because due to the effect of the transverse work force $F_r = F_S$ the screw body is exposed to shear due to its direct contact with the surface of the hole into which the screw is inserted ($D_o = d_s$, which means that the diameters of the hole and the screw body are exactly adjusted to one another). Here, the screw-nut is tightened slightly only to ensure securing of connected parts due to shifting, so that normal stress due to tightening do not exceed 20% of border stretching of the respective screw σ_{TM}:

$$\sigma_p = (F_p / A_3) \leq 0{,}2 \cdot \sigma_{TM}, \quad S_\sigma = (\sigma_{TM} / \sigma_p) \geq 5 \qquad (10)$$

S_σ – partial degree of security with respect to the normal stress.

3. APPEARANCE, DESCRIPTION AND PRINCIPLES OF WORK OF THE HYDROSTATIC PISTON DEVICE FOR CHECKING THE FORCE OF TIGHTENING AND SCREW DILATATION IN A SCREW CONNECTION

The scheme of hydrostatic piston device for measuring forces and dilatation of screw during tightening is shown in Fig. 7.

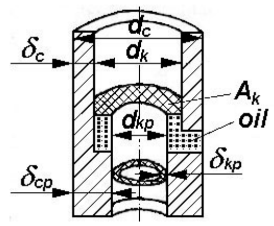

Fig. 7: *Simplified layout of device for measuring the force of tightening and screw dilatation in a screw connection*

The principle of operation of hydrostatic device for measuring the force of tightening and screw dilatation in a screw connection is based on the fact that due to the tightening of the screw-nut (or screw head) by means of wrench the screw becomes elastically elongated for the value $+\lambda_z$, which can be measured using a comparator (Fig. 8). In industrial practice, the use of optical comparators with a circular scale and a needle is the commonest (Fig. 9.a), while for laboratory and scientific measurements and testing in various institutes electronic comparators with digital reading are increasingly used (Fig. 9.b). Schematic layout of equipment for measuring the forces and dilatations of screw during tightening is shown in Fig. 10, on which only the manometer is not shown, which previously must be calibrated in decanewtons (*daN*) for the direct reading of the achieved tightening force (F_p), [8, 9].

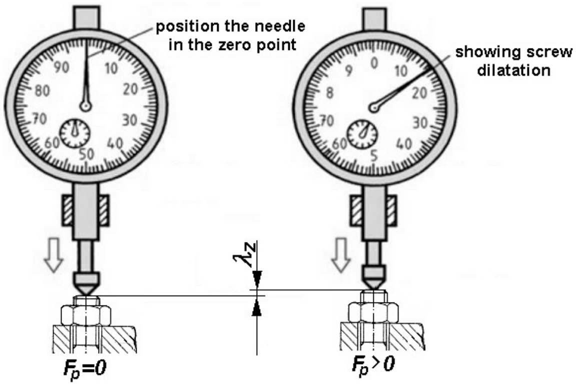

Fig. 8: *Measurement of dilatation of screw using comparator*

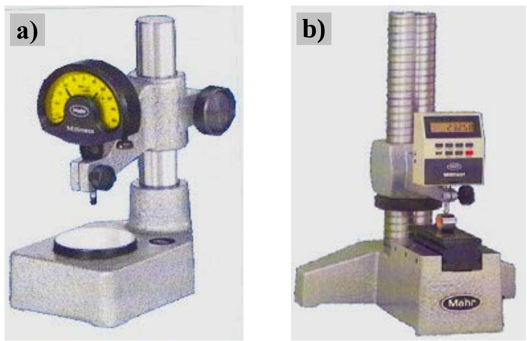

Fig. 9: *Optical comparator (a) and electronic comparator (b)*

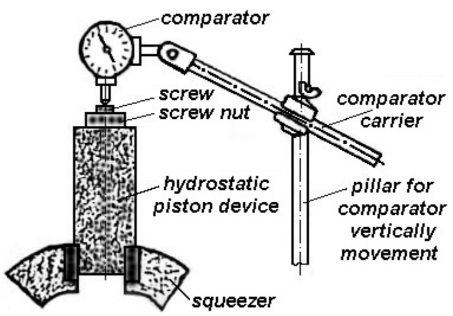

Fig. 10: *Schematic layout of equipment for measuring the force of tightening and screw dilatation in a screw connection*

The main part of the hydrostatic device is the cylinder whose wall thickness is δ_c, in whose interior there is a hollow piston with a piston extension, wherein δ_{cp} is the thickness of the cylinder wall at the point where the extension is (Fig. 7). The piston and the cylinder are mounted metal /on metal (without any seal). Since there is a very high pressure inside the cylinder, it is necessary to keep their touching surfaces finely processed, first by grinding and then by glazing. Therefore, the viscosity should be as large as possible so that the oil does not go through the gap between the piston and cylinder, and therefore the oil for shock absorbers is recommended because it is most suitable for these purposes.

Through the central hole of the hydrostatic device (through the cavity of the piston and the piston extension) the screw being tested is inserted (Fig. 11). In its lower part, the hole has hexagonal shape in which the screw head is placed in order to properly fix it, that is to prevent its turning when tightening. By pressing the screw-nut, the pressure on the piston head is caused, which will therefore tend to compress the oil in the cylinder. However, since the oil is incompressible, there is an elastic deformation in the screw in the form of its elongation $+\lambda_z$, while the resulting oil pressure, according to Pascal's law for liquids and incompressible fluids, is transmitted to the manometer, where it is directly read out as the tightening force of the screw F_p. It should be strictly taken into account that the device is well fixed in squeezer so as not to move during tightening, because this would inevitably lead to incorrect reading of the expansion of the screw on the comparator.

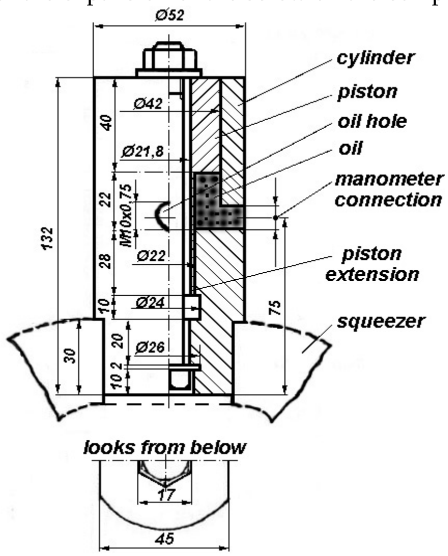

Fig. 11: *Measures and dimensions of the hydrostatic piston device for measuring the force of tightening and screw dilatation in a screw connection*

For this experiment, in principle, screw of any length and any diameter of its body may be used. It should be kept in mind that the screws of the smaller diameter and the lower class of strength are significantly more vulnerable than those with a larger diameter and a higher class of strength [10]. In the experiment, a screw with the middle class of strength of the following mark was selected:

SCREW M10x140-8.8 SRPS EN ISO 4016

In order to dimension the device, the initial assumption was that not only the most responsible screws are controlled, but also the screws with significantly weaker mechanical properties, which are in practice the most vulnerable to breakage. In them, tightening stress which is greater than the breaking point can be generated by tightening, so the starting idea is that the device should be able to produce a tightening force that will break the screw strength class 5.6, whose minimum tensile strength is $\sigma_M=R_m=500MPa$.

From the appropriate standards for metric thread M10 with a big step, the following values are found:
$d=10mm$- nominal thread diameter;
$d_2=9{,}026mm$- medium (theoretical) thread diameter;
$d_3=8{,}16mm$- diameter of screw core;
$A_3=(d_3^2\cdot\pi)/4=52{,}3mm^2$ - cross sectional area of the core of the screw;

P=1,5mm- thread pitch;
φ=3,028^o- angle of the rise of threads;
μ_n=0,15- slip friction coefficient in a threaded pair;
$\rho_n=arctg\mu_n=arctg0,15=8,83^o$- angle of friction in a threaded pair.

Maximum calculation value of the tightening force that would break the M10 screw with strength class 5.6 (which should be generated by hydrostatic device) is:

$$F_{pmax}=\sigma_M \cdot A_3=500\cdot 52,3=26150N=2615daN \quad (11)$$

Piston diameter (d_k) and piston extension diametar (d_{kp}) are empirically adopted, having in mind the fact that it is a screw of the nominal diameter M10:

$$d_k=42mm \quad (12)$$
$$d_{kp}=22mm \quad (13)$$

4. MANOMETER CALIBRATION PROCEDURE IN ORDER TO MEASURE THE FORCE OF TIGHTENING

It is necessary first to calculate the surface of the piston A_k which is double-shaded in Fig. 7, which represents the surface of the corresponding circular ring:

$$A_k=\frac{d_k^2\cdot\pi}{4}-\frac{d_{kp}^2\cdot\pi}{4}=\frac{42^2\cdot\pi}{4}-\frac{22^2\cdot\pi}{4}\approx 1000mm^2 \quad (14)$$

Maximum pressure in the cylinder during breaking the screw M10-5.6 would be:

$$p_{\max}=\frac{F_{p\max}}{A_k}=\frac{26150}{1000}\approx 26MPa\approx 260bar \quad (15)$$

Based on (15) it can be concluded that it is necessary to select a manometer that can measure pressures up to 30MPa, i.e. to 300bar.

Calibration of the manometer should be performed so that instead of each of the existing pressure values on manometer the corresponding value of the tightening force F_p must be entered, in accordance with the values given in the Table 3.

Table 3. *Corresponding pressure values and tightening forces values that are needed to calibrate the manometer*

Existing pressure value p [MPa] / ([bar])	Piston surface A_k [mm^2]	Force of tightening F_p [N]	Force of tighten. F_p [daN]
5 / (50)	1000	5000	500
10 / (100)	1000	10000	1000
15 / (150)	1000	15000	1500
20 / (200)	1000	20000	2000
25 / (250)	1000	25000	2500
30 / (300)	1000	30000	3000

This practically means that instead of reading the pressure of 50bar on the manometer the tightening force with value 500daN will be read, instead of reading the pressure of 100bar the tightening force with value 1000daN will be read, ... , instead of reading the pressure of 300bar the tightening force with value 3000daN will be read (Figure 12).

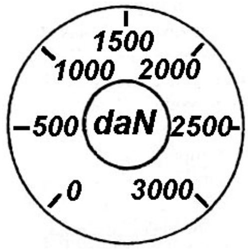

Fig. 12: *Manometer for measuring the tightening force of the screw*

When the maximum pressure value that the device can generate is known, the wall thickness of the hollow piston extension δ_{kp} can finally be determined as well as the thickness of the cylinder wall δ_c and δ_{cp} which are shown in Fig. 7:

$$\delta_{kp}=\frac{d_{kp}\cdot p_{\max}}{2\cdot\sigma_{zdoz}}=\frac{22\cdot 30}{2\cdot 210}=1,57mm$$

$$\delta_c=\frac{d_k\cdot p_{\max}}{2\cdot\sigma_{zdoz}}=\frac{42\cdot 30}{2\cdot 210}=3mm$$

σ_{zdoz}=210MPa- allowed value of the stress on the tensile strength of the structural steel Č0745 (material of piston and cylinder).

Given the recommended value of degree of security for thickness of the wall S=1,5...2 (it is a static strain), the following thicknesses of the walls can finally be adopted:

$$\delta_{kp}=3mm \quad (16)$$
$$\delta_c=5mm \quad (17)$$

The outer diameter of the cylinder (d_c):

$$d_c=d_k+2\delta_c=42+2\cdot 5=52mm \quad (18)$$

$$\delta_{cp}=\frac{d_c-d_{kp}}{2}=\frac{52-22}{2}=15mm \quad (19)$$

5. THEORETICAL VALUE OF THE FORCE OF TIGHTENING AND SCREW DILATATION

The main purpose of the performed experiments was to carry out the following tests:

- whether the theoretically calculated value of the tightening force F_{pt} corresponds to the actual value of the tightening force F_{ps} which was read on the manometer of the hydrostatic device;
- whether the theoretically calculated value of the dilatation of screw λ_{zt} corresponds to the actual value of the dilatation of screw λ_{zs} which was measured by means of comparator.

The theoretical value of the tightening force is obtained on the basis of the equation (2), bearing in mind that the other member on the right side of the equation is equal to zero (T_μ=0). The reason is that the frictional force during tightening cannot be transferred from the screw-nut to the head of the piston of the device, because the piston, together with the screw-nut, rotates freely around its vertical axis. In this way, the tightening moment will be equal to the moment necessary to overcome the friction between the screw threads and the screw-nut threads:

$$T_p = T_n = F_{pt} \cdot \frac{d_2}{2} \cdot tg(\varphi + \rho_n) = F_{pt} \cdot \frac{0{,}009026}{2} \cdot tg(3{,}028 + 8{,}53)$$

$$T_p = 0{,}000923 \cdot F_{pt} \quad [daNm] \qquad (20)$$

For the corresponding set value of torque of tightening T_p the theoretical value of the tightening force of the screw can be calculated using the following formula:

$$F_{pt} = \frac{T_p}{0{,}000923} = 1083{,}5 \cdot T_p \quad [daN] \qquad (21)$$

Theoretical value of dilatation of the screw λ_{zt} is calculated on the basis of the Hooke′s law, according to which the stress in the screw due to tightening σ_{zp} is proportional to the dilatation of the screw which can also be expressed by relative elongation ε_z:

$$\sigma_{zp} = E_z \cdot \varepsilon_z = E_z \cdot \frac{\lambda_{zt}}{l} = 2{,}1 \cdot 10^5 \cdot \frac{\lambda_{zt}}{120} = 1750 \cdot \lambda_{zt} \quad [MPa]$$

E_z=2,1·10^5MPa-modulus of elasticity of screw material (steel);

l≈120mm-the length of the screw between the screw head and the nut (thickness of the connected part, according to Fig. 11).

$$\lambda_{zt} = \frac{\sigma_{zp}}{1750} = \frac{F_{pt}}{A_3 \cdot 1750} = \frac{F_{pt}}{52{,}3 \cdot 1750} = 0{,}000011 \cdot F_{pt} \quad [mm]$$

where F_{pt} is expressed in Newton [N].

Since the manometer on the hydrostatic device is calibrated in the decanewton [daN], the following relationship will be applied:

$$\lambda_{zt} = 0{,}00011 \cdot F_{pt} = 1{,}1 \cdot 10^{-4} \cdot F_{pt} \quad [mm] \qquad (22)$$

6. TABULAR AND GRAPHICAL REVIEW OF THE MEASURED RESULTS

When tightening the screw-nut (or the screw head) loads are transmitted through contact partially from the threads of screw-nut (or from the threads of screw) to the threads of coupled element. They are the largest on the contact of the first active threads which are closest to the plates (in our case, the nearest piston of the hydrostatic cylinder) and they gradual decrease towards the upper threads, which are therefore considerably more relieved than the lower threads.

The experiment is performed by gradually increasing the intensity of tightening force F_p in a few selected steps, and our proposal is six steps because the manometer has the same number of assigned values (Fig. 12).

Calculated theoretical and measured real values of the tightening force are shown in Table 4, and calculated theoretical and measured actual values of dilatation of the screw are shown in Table 5.

Table 4. *Theoretical (calculated) values and actual (measured) values of the force of tightening of the screw*

Ordinal number of meas.	Tightening torque* T_p [$daNm$]	Tightening force** F_{pt} [daN]	Tightening force*** F_{ps} [daN]	Error $\|F_{pt} - F_{ps}\|$ [daN]	Error $\|F_{pt} - F_{ps}\|$ [%]
1	0,46	500	560	60	12
2	0,92	1000	880	120	13,6
3	1,38	1500	1450	50	3,4
4	1,85	2000	2100	100	5
5	2,31	2500	2350	150	6,4
6	2,77	3000	2750	250	9,1

* Based on the formula (20) in theoretical calculations, and when measured it is read on the torque scale;
** Based on the formula (21);
*** It is read on the manometer of the hydrostatic device.

Table 5. *Theoretical (calculated) values and actual (measured) values of screw dilatation during tightening*

Ordinal number of meas.	Tightening force* F_p [daN]	Screw dilatation** λ_{zt} [mm]	Screw dilatation*** λ_{zs} [mm]	Error $\|\lambda_{zt} - \lambda_{zs}\|$ [mm]	Error $\|\lambda_{zt} - \lambda_{zs}\|$ [%]
1	500	0,055	0,040	0,015	37,5
2	1000	0,110	0,130	0,020	18,2
3	1500	0,165	0,180	0,015	9,1
4	2000	0,220	0,190	0,030	15,8
5	2500	0,275	0,290	0,015	5,4
6	3000	0,330	0,360	0,030	9,1

* When measured it is read on manometer of hydrostatic device;
** Based on formula (22);
*** It is read on comparator.

The obtained results from Tables 4 and 5 can be graphically displayed by the deformation diagram of the screw connection (Fig. 13). On the ordinate axis of this diagram, the values of the tightening force are shown in daN, and on the abscissa axis the screw dilatations are shown in mm. By connecting points with coordinates [λ_{zt} ; F_{pt}] for each concrete measurement, a straight line is obtained which shows the theoretical deformation of the screw. By analogue connection of the points with coordinates [λ_{zs}; F_{ps}] for each specific measurement the straight line is also obtained which shows the actual deformation of the screw during its tightening.

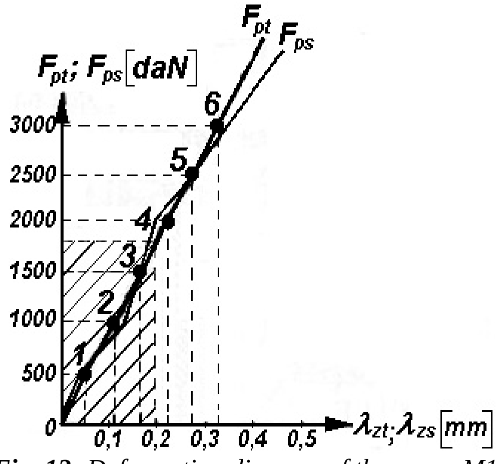

Fig. 13: *Deformation diagram of the screw M10 strength class 8.8 during tihtening*

7. CONCLUSION

The previously described method of measuring the force of the tightening and dilatation of the screw was developed at the High Technical Mechanical School of Applied Studies in Trstenik. The same can be applied in practice as an original laboratory-experimental method. With it, in a relatively simple and fast manner, the screws of various sizes and different strength classes can be checked using the hydrostatic device shown in Fig. 11, in all types of prestressed screw connections. Such testing in practice can be of multiple significance [11, 12]:

• To demonstrate to students in technical schools and faculties [13] the importance of correct selection of the tightening force, which must always be within the given range [14, 15]:

$$F_{pmin} \leq F_p \leq F_{pmax} \tag{23}$$

If the screw connection is insufficiently tightened ($F_p < F_{pmin}$) splitting of connected parts can occur during working life. Otherwise, if it is too tightened ($F_p > F_{pmax}$) permanent i.e. plastic deformations may occur either in the screw, or in the other elements of the screw connection. Therefore, the actual value of the tightening force can only be found in the elastic area, and that means only in a shaded area on the deformation diagram of the screw shown in Fig. 13.

• In industry, the mentioned method enables to compare the screws of the same size and the same class of strength, but from different manufacturers. First it is necessary to carry out proper sampling, i.e. selection for the purpose of analyzing the optimal number of screws from each manufacturer individually, based on the size of the series of purchased screws. This can quickly identify manufacturers whose screws are of significantly lower quality than the standard quality, as well as producers whose quality may be higher than the usual (average) quality on the market [16, 17, 18, 19].

• In research laboratories and institutes it is possible to devise completely new and original experiments, which aim at deeper research of some specific problems related to work and realization of screw connections. Thus it is possible to research: the influence of increased (or reduced) temperatures on the dilatations of the screw during tightening; the influence of threads step (fine, rough) on the forces and dilatations of the screw; influence of the length of carrying the threaded joint (the numbers of screw threads and the screw-nut threads in active contact) to the forces and dilatations in the screw; the influence of vibration during tightening on the forces and deformations in the screw, etc. [20, 21, 22].

Finally, it should be noted that it is possible, in addition to M10 screw, by means of the above method, to test the screws having larger or smaller nominal diameters and larger or smaller lengths than those indicated in the experiment. The only difference is that in this case the measures and dimensions of the hydrostatic device shown in Figure 11 should be adjusted to the diameter and length of the screw being examined, while the principle of measuring the forces and dilatations, as well as calibration of the manometer, would remain unchanged.

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