Specifičnosti procesa namotavanja trakastog materijala u obliku asure od sečenih staklenih vlakana (MAT-a)

Milutin Živković^{1,*}, Jasmina Milojković², Ivan Bijelić³

¹ Visoka tehnička mašinska škola strukovnih studija, Trstenik, Srbija

² Fakultet inženjerskih nauka u Kragujevcu, Kragujevac, Srbija

³ Viša tehnička mašinska škola strukovnih studija – Novi Beograd, Srbija

Rezime: Procene namotavanja (odmotavanja) karakterišu promenljivi kinematski i dinamički parametri. Tako, u slučaju povećanja prečnika namotanog materijala smanjuje se frekvencija njegovog obrtanja i obrnuto. Izvedeni sistem je specifičan i po načinu prenosa obrtnog momenta, frikcionim prenosom. Iz tih razloga, u radu su na primeru namotavanja asure od sečenih staklenih vlakana, proizvodnje ETEKS - a iz Baljevca, matematički definisani promenljivi kinematski i dinamički pogonski parametri. Da bi se proces namotavanja odvijao konstantnom vučnom silom i linijskom brzinom, snaga motora na vučnom valjku (namotavanju) mora biti konstantna. Ona mora biti takva da obezbeđuje proklizavanje unutrašnjih među slojeva ali da ne dođe do prekida (kidanja) osnovnog materijala. Dobijene zavisnosti su simulirane na računaru, za asuru MAT-a gramature 450 g/m² a dobijeni rezultati pokazali su značajnu saglasnost sa rezultatima težinskog merenja namotanih asura projektovanog uređaja u radu.

Ključne reči: namotavanje (odmotavanje), staklena vlakna, matematički model, promenljivi parametri, simulacija, kartonska kutija, težinsko merenje namotanih rolni.

0. UVOD

DP industrija staklenih vlakana "ETEX" iz Baljevca proizvodi jebezalkalna staklena vlakna (MAT i ROVING), koja su mineralnog porekla, poznato pod internacionalnom oznakom "E", što podrazumeva tačno određen hemijski sastav staklasl.1[1]. Instalirani godišnji kapacitet je oko 5400 tona gotovih proizvoda u obliku MAT-a i ROVING-a (odnos: 40% MAT, 60% ROVING). Ista je zapošljavala 400 radnika, a izrađena je po licenci Engleske firme "TBA – BISHOP" [1,2]. Međutim, njena sudbina je slična sa sudbinom mnogih naših proizvodnih firmi, zatvorena je krajem 2010. god.

Sl. 1 Detalj iz proizvodnje staklenih vlakana

MAT se proizvodi u obliku asure od sečenih staklenih vlakana, koja su međusobno povezana odgovarajućim vezivnim sredstvom. Proizvodi se kao emulzioni M1102 (inter. oznaka) i praškasti M1104, širine 1000 do 3000 (mm), namotani na kartonske rolne do 110 (m) dužine. Nakon namotavanja iste se pakuju u kartonske kutije, dimenzija 300 x 300 [mm], a iste se

zatim paletiraju (18 kutija). Gramatura MAT-a kreće u granicama od 250 do 900 gr/m² [1,2].

U radu je objašnjena specifičnost prenosa obrtnog momenta kao i način namotavanja njenom izlaznom delu proizvodne linije sl. 2, [1,2]. Takođe, objašnjene su sve specifičnosti koje se odnose na namotavanje ovakvih materijala.

Sl. 2 Izgled izlaznog dela linije za namotavanje

Vrsta materijala namotavanja (čelične ili bakarne trake, Al - folije, tekstilni materijali ili hartija) utiče na izbor načina pogona, zavisno kako od mehaničkih tako i od tehnoloških karakteristika. Iz iskustva autora ovog rada kao i dostupnih literaturnih podataka poznato je da se za namotavanje (odmotavanje) ovakvih materijale koriste elektromotorni i hidraulički pogoni [2,3]. Pored načina pogona, kod ovakvih sistema, mora se voditi računa i o bočnom zakrivljenju ivica i mogućeg razloga za oštećenja namotavanog materijala tj. regulacija ivica [2,4,5,6,7].

Zato se pri izboru načina pogona mora sprovesti matematička analiza i provera dinamičkih i statičkih

parametara kao i konstruktivna upoređenjakako bi se izabralo optimalno rešenje [2,4].

Osnovni uslov, koji se traži od svih uređaja za namotavanje ili odmotavanje, je konstantna linijska brzina kao i sila zatezanja u materijalu. Sam izbor upravljačkog uređaja vrši se prema kriterijumu, koji treba da zadovoljiti postavljene zahteve vodeći računa o promenljivim kinematskim i dinamičkim parametrima namotavanja. Kako je problem namotavanja asure od sečenih staklenih vlakana složen i zbog načina prenosa obrtnog momenta (frikcionim prenosom) to rešenje, za napred navedene probleme, čini još složenijim.

Dobijene matematičke zavisnosti su proverene na simulacionom modelu a dobijeni rezultati bi trebali da su saglasni sa stvarno izmerenim težinama na uređaju koji je redizajniran [2,3,8].

1. PRENOSA OBRTNOG MOMENTA KOD NAMOTAVANJA ASURE OD MATA - A

Ugaona brzina i obrtni moment pogonske mašine retko se poklapaju sa zahtevima radne mašine. Najčešće korišćeni asinhroni motori naizmenične struje imaju 2,4 i 6 polova, a kod frekvencije $f = 50\text{Hz}$ brojevi obrtaja ovih motora su 3000, 1500 ili 750 min^{-1} . Motori sa unutrašnjim sagorevanjem takođe ekonomično rade u uskom području učestanosti obrtaja. Radne mašine pak najčešće zahtevaju velike obrtne momente pri malim brzinama tako da je neophodna ugradnja prenosnika snage između pogonske i radne mašine [2,9,10,11,12].

Zadatak prenosnika je pretvaranje i "vođenje" mehaničke energije od izlaznog vratila pogonske mašine (motor, turbina) do mesta gde je ona neophodna – ulazno vratilo radne mašine, odnosno pogonski točkovi mobilnog sredstva. Zavisno od uslova, koje diktiraju radne mašine, razlikuju se prenosnici sa konstantnim i prenosnici sa promenljivim prenosnim odnosom. U principu se može reći da se prenosnik sastoji iz tri dela – ulaznog i izlaznog vratila kao i kućišta, u kome su vratila na osnovu mehaničkog, električnog hidrauličkog ili pneumatskog principa međusobno povezana. Ako se veza između pogonskog i gonjenog vratila bazira na mehaničkom principu, onda se radi o mehaničkim prenosnicima. Oni prenose energiju odnosno obrtni momenat i ugaonu brzinu na dva osnovna načina: oblikom i trenjem (prljanjanjem). Prenos može biti ostvaren neposrednim dodirivanjem pogonskog i gonjenog elementa, ili preko posrednika.

Obrtni moment se između pogonskog i gornjeg vratila može preneti i trenjem tj. neposrednim dodirivanjem. Pri okretanju pogonskog frikcionog točka na dodirnoj površini točkova javlja se sila trenja, usled koje dolazi do okretanja i drugog – gonjenog frikcionog točka. Da bi rad frikcionog prenosnika bio moguć potrebno je da sila trenja (F_{μ}) bude uvek veća, ili u graničnom slučaju jednaka tangennoj tj. obinnoj sili (F_t) na frikcionim točkovima, izraz (1):

$$F_{\mu} \geq F_t \quad (1)$$

Potrebna sila trenja, na dodirnim površinama frikcionih točkova, dobija se uzajamnim pritiskom jednog frikcionog točka na drugi. Sve prednosti ovakvih prenosnika su dobro poznate i dobro obrađene u stručnoj literaturi [4,5]. Izbor materijala frikcionih parova igra

važnu ulogu za nosivost i radni vek prenosnika. On treba da ispuni sledeće zahteve:

- Veliku otpornost u odnosu na dodirni pritisak i u odnosu na habanje, da bi bio obezbeđen dovoljan radni vek prenosnika (kaljen čelik);
- Veliki koeficijent trenja, da bi za zadatu silu pritiska bila obezbeđena velika sila trenja (guma),
- Veliki modul elastičnosti, da bi se smanjile elastične deformacije, samim tim i elastično klizanje.

Oblik radnih površina frikcionih parova tesno je povezan sa odnosom ugaonih brzina kao i međusobnim položajem osa obrtanja frikcionih točkova. Naime, on mora da zadovolji uslov kotrljanja bez klizanja jedne radne površine u odnosu na drugu, što znači da obimne brzine dodirnih tačaka moraju biti jednake sl. 3[2,9,10]. Veličine, koje određuju položaj jedne ose obrtanja u odnosu na drugu, kao i odnos intenziteta odgovarajućih ugaonih brzina su: osno rastojanje i kinematski prenosni odnos.

Sl. 3 Namotavanje frikcionim kontaktom para za prenos obrtnog momenta (namotavanje MAT-a)

Oсно rastojanje je najkraće rastojanje osa obrtanja (min i max) dato je izrazom (2) [2]:

$$\frac{D_{A+d}}{2} = a_1 \leq a_2 \leq \frac{D_{A+D}}{2} = a_{\max} \quad (2) \text{ Obimna}$$

$$v = \frac{D_A}{2} \cdot \omega_1 = \text{const} \quad (\text{za frikcionirani točak 1}) \quad (3)$$

$$V_1 = \frac{d_2}{2} \cdot \omega_2 = \frac{2 \cdot r_2}{2} \cdot \omega_2 = r_2 \cdot \omega_2 \quad (4)$$

Može se izračunati nominalni kinematski odnos (i_0), kao odnos inteziteta vektora ugaonih brzina. On je uvek pozitivna veličina, veća ili jednaka jedinici (gde važi jednakost brzina tj. $v_1=v_2=v=\text{const}$), izraz (5) [2,9,10].

$$i_0 = \frac{\omega_1}{\omega_2} = \frac{\frac{d_2}{2} \cdot v_2}{\frac{D_A}{2} \cdot v_1} = \frac{d_2}{D_A} \quad (5)$$

Zbog elastičnog klizanja između dodirnih površina frikcionog para, stvarni prenosni odnos (i) odstupa u odnosu na normalni, izraz (6) [2,9,10]:

$$i = \frac{i_0}{1 - \epsilon} \quad (6)$$

Da bi se mogao, frikcionim prenosom, preneti obrtni moment potreban i dovoljan uslov dat je izrazom (1) a mehanizam prenosa pokazuje sl. 4. Na sl. 5 su sile razložene za pogonski točak i za doboš namotavanja [2].

Sl. 4 Sile na mestu dodira frikcionog para

Na osnovu izraza (1) kao i sl.5 dobija se izraz (7) tj. granični slučaj [2,9,10]:

$$F_N \cdot \mu = S_\mu \cdot F_t \quad (7)$$

Gde je: F_N [kN] – normalna sila, (ostvarena je na mehanizmu kolica pomoću pneumatskih cilindara a zavisi od pritiska vazduha u pneumatskom sistemu), μ – koeficijent trenja staklenih vlakana na mestu dodira, S_μ – stepen sigurnosti protiv klizanja iznosi 1,5 ... 2 (za tačan oblik aksiode bez kinematskog klizanja) [2,9,10].

Sl. 5 Sile na mestu dodira frikcionog para

F_t [kN] – tangentna sila na obodu pogonskog točka, koja se može izračunati pomoću izraza (8) [2,9]:

$$F_t = \frac{T_1}{D_A} = \frac{2 \cdot T_1}{D_A} \quad (8)$$

Shodno izraza (8), za prenos potrebne obimne sile, potrebna normalna sila između dodirnih površina može se iskazati izrazom (9):

$$F_N = \frac{S_\mu \cdot F_t}{\mu} = 2 \cdot \frac{T_1 \cdot S_\mu}{\mu \cdot D_A} \quad (9)$$

Ona je stalna za svo vreme procesa namotavanja.

2. MEHANIKE NAMOTAVANJA SAVITLJIVIH MATERIJALA

Mnogi tehnološki procesi, u oblasti "dorade" (tkanine i papir) kao i smanjenja debljine (folije, čelični limovi i limovi od obojenih materijala) zahtevaju premotavanje materijala sa jednog valjka na drugi kao i obratno. Proces zahteva, da je linijska (transportna) brzina "dorađivanog" materijala konstantna za svo vreme namotavanja. Problem namotavanja se sastoji u što tačnijem matematičkom formulisanoj i uspostavljanju kvantitativne i ligičke zavisnosti pogonskih parametara.

Pored klasičnih mehaničkih sistema i hidrostatika je našla svoje mesto u rešavanju ovih problema razvijenih komponente koje su joj to i omogućile. U novije vreme standardni elektromotorni pogoni sve više nalaze svoje mesto u rešavanju ovih problema zahvaljujući razvoju elemenata upravljačke elektronike [2,3,4,5,6,7,11,12]. Za rešavanje ovih problema, nezavisno od vrste pogona,

neophodno je prvo odrediti sve geometrijske, kinematske kao i dinamičke funkcionalne veličine.

2.1 Geometrija sistema

Na osnovu jednakosti zapremine materijala, pre i posle namotavanja na doboše (valjke), shodno sl.6 može se uspostaviti zavisnost izrazom (10) [2,3,7]:

$$L \cdot \delta \cdot b = (D^2 - d^2) \cdot \frac{\pi}{4} \cdot b \quad (10)$$

gde je: L [m] – dužina materijala koji se namotava, δ [m] – debljina materijala koji se namotava, b [m] – širina materijala koji se namotava, $D=2R$ [m] – prečnik namotanog valjka $d=2r$ [m] – spoljni prečnik praznog (nenamotanog) valjka.

Sl.6 Opšti slučaj premotavanja trakastog materijala sa jednog valjka na drugi

Na osnovu izraza (10), dužina namotanog materijala, kao funkcija poluprečnika namotanog valjka i debljine materijala namotavanja, data je izrazom (11) [2]:

$$L = (R^2 - r^2) \cdot \pi / \delta \quad (11)$$

Izraz (11) može se ilustrovati i u vidu dijagrama prikazanog na sl.7 [2].

Sl. 7 Zavisnost dužine i debljine namotanog materijala za konstantni prečnik namotavanja D

Ako posmatramo opšti slučaj, da je deo materijala premotan sa valjka 1 na valjak 2, kao na sl.6, iz uslova jednakosti dužina dobija se izraz (12):

$$L = L_1 + L_2 \quad (12)$$

gde je: L_1 [m] – dužina materijala na valjku 1, L_2 [m] – dužina materijala na valjku 2.

Na osnovu izraza datog izrazom (12) može se dobiti izraz (13):

$$(R^2 - r^2) \cdot \frac{\pi}{\delta} = (r_1^2 - r^2) \cdot \frac{\pi}{\delta} + (r_2^2 - r^2) \cdot \frac{\pi}{\delta} \quad (13)$$

Posle sređivanja dobija se izraz (14):

$$r_1^2 + r_2^2 = R^2 + r^2 = \text{const} \quad (14)$$

Iz izraza (14) može se zaključiti da se zavisnost tekućih parametara menja prema jednačini kruga, što se može ilustrovati sl. 8, [2].

Sl. 8 Promene "tekućih" prečnika namotavanja (odmotavanja)

2.2 Kinematika sistema

Određivanje kinematskih veličina posmatraćemo kako za "vučni valjak" (valjak na koji se željeni materijal namotava) tako i za "vučeni valjak" (valjak sa koga se odmotava materijal). Prema sl. 6 vučni valjak obeležen je brojem 2, vučeni sa 1, [2].

Vučni valjak: Iz uslova jednakosti linijske brzina namotavanog materijala ($v = \frac{dL_2}{dt} = \text{const}$), dužina L_2 se može odrediti shodno izrazu (15):

$$L_2 = v \cdot t \quad (15)$$

Gde je: v [m/sec] – linijska brzina (obimna brzina vučnog valjka) a t [s] – vreme.

Zamenom izraza (15) u izraz (11) dobija se izraz (16):

$$v \cdot t = (r_2^2 - r^2) \cdot \frac{\pi}{\delta} \quad (16)$$

Shodno izrazu (16) promenljivi parametar (r_2) se može izraziti jednačinom (17) [2]:

$$r_2 = \sqrt{r^2 + \frac{\pi}{\delta} \cdot v \cdot t} \quad (17)$$

Izraz (17) pokazuje promenu poluprečnika vučnog valjka u funkciji vremena, ali kako je obimna brzina konstantna, tj. njena promena biće izražena sa (18):

$$\omega_2 = \frac{v}{r_2} \quad (18)$$

gde je: ω_2 [s⁻¹] – ugaona brzina vučnog valjka

Kada je poznata promena ugaone brzine vučnog valjka (doboša), može se odrediti broj obrtaja vučnog valjka, na osnovu dobro poznatog izraza iz mehanike za ugaonu brzinu ($\omega = \frac{n \cdot \pi}{30}$). Broj obrtaja vučnog valjka (n_2) može se izračunati shodno izrazu (19) [2]:

$$n_2 = \frac{30 \cdot v}{\pi \cdot r_2} \quad (19)$$

Linijska brzina je konstantna (projektni zahtev) i ona za vučeni doboš se može odrediti prema izrazu (20):

$$v = r_2 \cdot \omega_2 = \text{const} \quad (20)$$

Diferenciranjem izraza (20) po vremenu dobija se izraz dat jednačinom (21):

$$\frac{dv}{dt} = \frac{dr_2}{dt} \cdot \omega_2 + r_2 \cdot \frac{d\omega_2}{dt} = 0 \quad (21)$$

Kako je $\frac{dr_2}{dt} = \frac{\delta \cdot v}{2\pi \cdot r_2}$ dobija se izraz (22) [2]:

$$\frac{d\omega_2}{dt} = \dot{\omega}_2 = \frac{\delta \cdot v^2}{2\pi \cdot r_2^3} \quad (22)$$

gde je: $\dot{\omega}_2$ [sec⁻²] – ugaono ubrzanje vučnog valjka.

Vučeni valjak: Preostala količina materijala, na vučenom valjku, u posmatranom trenutku može se odrediti shodno izrazima (23 i 24) [2]:

$$L_1 = L - L_2 \quad (23)$$

$$(r_1^2 - r^2) \cdot \frac{\pi}{\delta} = (R^2 - r^2) \cdot \frac{\pi}{\delta} - v \cdot t \quad (24)$$

Rešavanjem jednačine (24) po nepoznatoj (r_1) dobija jednačina data u obliku izraza (25):

$$r_1 = \sqrt{R^2 - \frac{\delta}{\pi} \cdot v \cdot t} \quad (25)$$

Izraz (25) pokazuje promenu poluprečnika vučnog valjka u funkciji vremena. Ako ponovimo kompletan tok proračuna kao za vučni valjak dobijaju se sledeći izrazi (26, 27 i 29) [2]:

$$\text{Ugaona brzina vučnog valjka: } \omega_1 = \frac{v}{r_1} \text{ [s}^{-1}\text{]} \quad (26)$$

$$\text{Broj obrtaja vučnog valjka: } n_1 = \frac{30 \cdot v}{\pi \cdot r_1} \text{ [min}^{-1}\text{]} \quad (27)$$

$$\text{Ugaono ubrzanje: } \dot{\omega}_1 = \frac{d\omega_1}{dt} = \frac{\delta \cdot v^2}{2\pi \cdot r_1^3} \text{ [s}^{-2}\text{]} \quad (28)$$

S obzirom da se prenos obrtnog momenta i pogon "vučnog valjka" (na koji se namotavaju trake od staklenih vlakana) ostvaruje frikcionim prenosom to važe sve izvedene jednačine kao za vučni valjak, što možemo ilustrovati sl. 9 [2].

$$\omega_2 = \frac{v}{\sqrt{r^2 + \frac{\delta}{\pi} \cdot v \cdot t}} \quad (29)$$

Na osnovu izraza (18), gde je linijska brzina svo vreme rada $v = \text{const}$, dobija se (29) koji daje zavisnost ugaone brzine (broja obrtaja) namotavanog materijala od prečnika namotavanja (r_2) [2].

Sl. 9 Kinematika namotavanja staklenih vlakana (MAT-a)

2.3 Dinamika sistema namotavanja

Dinamika pogona uređaja za namotavanje trakastih materijala spada u domen problematike mehanizama sa promenljivom masom. Projektovani mehanizam namotavanja sa proklizavanjem međuslojeva, što omogućuje eliminisanje pojave labavog namotavanja, šematski je prikazan sl. 10 [2].

Na osnovu toga je i projektovan mehanizam za namotavanje sa proklizavanjem unutrašnjih međuslojeva tj. eliminisanje ugrađenog „labavog“ namotavanja [1,2].

Sl. 10 Kinematska šema mehanizma za zatezanje proklizavanjem unutrašnjih međuslojeva (tvrdo namotavanje)

Kako je debljina trake trakastog materijala staklenih vlakana MAT-a mala u odnosu na početni prečnik nenamotane kartonske cevi ($\phi d \gg \delta$), to se može smatrati da se težišna osa namotača poklapa sa osom obrtanja uređaja bez trake.

Masa valjka (m) u nekom proizvoljnom trenutku vremena namotavanja data je izrazima (30 i 30a) [2]:

$$m = m_0 + m_2 \quad (30)$$

$$m = m_0 + L_2 \cdot b \cdot \delta \cdot \rho \quad (30a)$$

Gde je: L_2 [m] – dužina namotanog materijala u nekom trenutku vremena,

m_2 [kg] – masa namotanog materijala u funkciji poluprečnika namotavanja r_2 (funkcija vremena t),

$m_0=30$ [kg] – masa praznog valjka bez namotanog materijala,

δ [m] – debljina namotavanog materijala,

ρ [kg/m³] – gustina materijala,

b [m] – širina namotanog materijala.

Konstruktivno se može podesiti da masa praznog valjka ima vrednost ($m_0 = \pi \cdot b \cdot \rho \cdot r^2$) pa se identički može pisati i izraz (31): $m_2 = \pi \cdot b \cdot \rho \cdot r_2^2$ (31)

Moment koji potiče od inercije valjka i namotanog materijala je dat izrazom (32):

$$M_{i2} = J_2 \cdot \frac{d\omega_2}{dt} = \frac{m_2 r_2^2}{2} \cdot \frac{d\omega_2}{dt} \quad (32)$$

Na osnovu izraza (3) i (22) i zamene u (32) dobija se izraz (33.1 i 33.2) [2,12,13,14]:

$$M_{i2} = \frac{\pi \cdot b \cdot \rho \cdot D_2^2}{2} \cdot r_2^2 \cdot \frac{d\omega_2}{dt} \quad (33.1)$$

$$M_{i2} = \frac{\pi \cdot b \cdot \rho}{2} \cdot \left(\frac{v}{\omega_2} \right)^4 \cdot \frac{d\omega_2}{dt} \quad (33.2)$$

Za ugaonu brzinu namotavanog valjka možemo napisati izraz (34):

$$\omega_2 = \frac{\omega_m}{i} = \frac{\omega_m}{i_f \cdot u_1} \quad (34)$$

Gde je ekvivalentni prenosni odnos posnika:

$$i = i_f \cdot u_1 = 47,84 \cdot 0,912 = 43,63 \quad (35)$$

Na osnovu izraza (32), (33), (34) i (35) može se izrazom (36) uspostaviti veza momenta namotavanja i momenta na osovini pogonskoj osovini motora, izrazi (36.1, 36.2 i 36.1) [2,12,13,14,16,17,18]:

$$M_{ieM} = \frac{M_{i2}}{i} = \frac{\pi \cdot b \cdot \rho \cdot v^4 \cdot i^2}{2} \cdot \frac{1}{\omega_m^4} \cdot \frac{d\omega_m}{dt} \quad (36.1)$$

$$M_{i2} = \frac{\pi \cdot b \cdot \rho \cdot v^4 \cdot i^3}{2} \cdot \frac{1}{i} \cdot \frac{d\omega_m}{dt} \quad (36.2)$$

$$M_{i2} = \frac{\pi \cdot b \cdot \rho \cdot v^4 \cdot i^2}{2} \cdot \frac{d\omega_m}{dt} \quad (36.3)$$

Sa druge strane, jednačina dinamička ravnoteže svedena na vratilo pogonskog elektromotora može se izraziti korišćenjem izraza (37) [2,]:

$$M_e = J_m \cdot \dot{\omega}_m + K_t \cdot \omega_m + M_{ieM} \quad (37)$$

Gde je: J_m [kg · m²] – moment inercije vratila elektromotora kao i svih njegovih obrtnih delova,

K_t – koeficijent trenja,

M_e – elektromagnetni momenat,

M_{ieM} – moment sveden na osovini motora.

Na osnovu izraza (36.1) i (37) može se dobiti izraz (38) [2,6,7,12,13]:

$$M_{ieM} = \left(J_m + K_{Me} \cdot \frac{1}{\omega_m^4} \right) \cdot \frac{d\omega_m}{dt} + K_t \cdot \omega_m \quad (38)$$

$$K_{Me} = \frac{\pi \cdot b \cdot v^4}{2} \cdot i^4$$

Za električni deo elektromotora kao i naponskog invertora sa strujnom regulacijom, izraz (39) [2,12,13]:

$$M_e = K_{inv} \cdot \omega_{mc} \quad (39)$$

Gde je: K_{inv} – konstanta invertora (zajedno sa električnim delom el. motora),

ω_{mc} – referentna brzina elektromotora.

Shodno jednakosti elektromagnetnih momenata, datih izrazima (38) i (39), dobijen je konačan izgled izraza za simulaciju (40):

$$\left(J_m + K_{Me} \cdot \frac{1}{\omega_m^4} \right) \cdot \frac{d\omega_m}{dt} + K_t \cdot \omega_m = K_{inv} \cdot \omega_{mc} \quad (40)$$

Dobijeni izraz može se predstaviti pomoću strukturnog blok dijagrama datog na sl. 6 [2,12,13].

Sl. 11 Blok dijagram uređaja za namotavanje sa proklizavanjem međuslojeva (4.22)

Da bi obimna brzina valjka bila konstantna potreban uslov je da referentna ugaona brzina (ω_{mc}) tj. brzina motora (u vremenu) treba da se menja po zakonu datim izrazom (41) [2,12,13]:

$$\omega_{mc} = \frac{v \cdot t}{\sqrt{r^2 + \frac{\rho \cdot v}{\pi} \cdot t}} \quad (41)$$

4. REZULTATI SIMULACIJE

Simulacijom izraza (40), (4.25), predstavljenog u obliku strukturnog blok dijagrama na sl. 11, na računaru jesu u programskom jeziku "MATLAB" simulirani parametri regulacije. Kao pomoć su korišćene simulacione šeme sa sl.12 kao i kinematska šema sa sl.10 [2]. Provera kinematskih i dinamičkih parametara izvršena za MAT, gramatureu ETEX-su, od 450 [gr/m²] a pomoćni podaci za proračun su [2]:

$$r = \frac{d}{2} = \frac{100}{2} = 50 \text{ [mm]} - \text{poluprečnik ne namotane kartonske cevi;}$$

$$R = \frac{D}{2} = \frac{300}{2} = 150 \text{ [mm]} - \text{poluprečnik}$$

namotanog materijala;

$L = 85 \cdot 10^3 \text{ [mm]}$ – zadata optimalna dužina trake MAT-a;

$b = 2500 \text{ [mm]}$ – širina trake koja se namotava ($2 \times 1250 \text{ mm}$);

$v \sim 9 \text{ [m/min]}$ – linijska brzina trake, na izlazu iz sušare tj. na uređaju za namotavanje;

$m_0 = 30 \text{ [kg]}$ – masa obrtnih delova namotača trake.

Promena ugaone brzine (ω_m), na osovini elektromotora (ulaz u reduktor), a prema karakteristikama ugrađenog sistema pokazan je na kinematskoj šemi sl. 10. Električni moment motora (elektromagnetni ili moment konverzije) dobija se kada se mehanička izlazna snaga motora podeli sa ugaonom brzinom rotora ω_m , što pokazuju izračunate vrednosti [2]:

$$\omega_m = \frac{\pi \cdot n_m}{30} = \frac{\pi \cdot 1380}{30} = 144,5 \text{ [s}^{-1}\text{]}$$

$$T_m = \frac{P_m}{\omega_m} = \frac{0,55 \cdot 10^3}{144,5} = 3,81 \text{ [N m]}$$

$$T_{nu_{max}} = T_m \cdot i_r \cdot u_l$$

$$T_{nu_{max}} = 3,81 \cdot 47,84 \cdot 0,912 \approx 166,2 \text{ Nm}$$

Ukupna površina trake:

$$L \cdot b = 85 \text{ [m]} \cdot 2,5 \text{ [m]} = 212,5 \text{ m}^2$$

- Ukupna masa namotane trake je:

$$m = \rho_p \cdot L \cdot b = 0,450 \text{ [kg} \cdot \text{m}^{-2}\text{]} \cdot 212,5 \text{ [m}^2\text{]}$$

$$m = 95,625 \text{ [kg]}$$

- Zapremina namotane trake:

$$V = L \cdot b \cdot \rho = 85 \text{ [m]} \cdot 2,5 \text{ [m]} \cdot 0,74 \cdot 10^{-3} \text{ [m]}$$

$$V = 0,157 \text{ [m}^3\text{]}$$

- Debljinu trake dobićemo izrazom (2):

$$\delta = \frac{(R^2 - r^2)}{L} \cdot \pi = \frac{(150^2 - 50^2)}{85 \cdot 10^3} \cdot \pi = 0,74 \cdot 10^{-3} \text{ m}$$

- Zapreminska gustina:

$$\rho = \frac{m}{V} = \frac{\rho_p \cdot L \cdot b}{L \cdot b \cdot \delta} = \frac{\rho_p}{\delta} = \frac{0,45}{0,74 \cdot 10^{-3}} = 608,11 \text{ kg/m}^3$$

Prema jednačini (2) dobija zamenom promena (povećanje) mase u funkciji vremena (t), izraz 21a se može izraziti novim izrazom (42) [2].

$$m = 30 + 0,15 \cdot 2,5 \cdot 0,74 \cdot 10^{-3} \cdot 608,11 \cdot t$$

$$m = 30 + 0,169 \cdot t \text{ [kg]} \quad (42)$$

Promena prečnika namotavanja, prema izrazu (8), gde se zamenom izračunatih vrednosti dobija njegova vremenska promena, izraz (43):

$$r_2 = \sqrt{r^2 + \frac{\delta}{\pi} \cdot v \cdot t} = \sqrt{50^2 + \frac{0,74}{\pi} \cdot 150 \cdot t}$$

$$r_2 = \sqrt{2500 + 35,332 \cdot t} \text{ [mm]} \quad (43)$$

Korišćenjem izraza (43) može se izračunati ukupno vreme namotavanja, izraz (44):

$$t = \frac{r_2^2 - 2500}{35,332} \quad (44)$$

U slučaju završetka namotavanja ($r_2 = R$) može se odrediti, shodno izrazu (36), ukupno vreme namotavanja (t_2). Maksimalni prečnik namotavanja mora da zadovolji uslov njihovog smeštaja u kutije standardnih dimenzija $300 \times 300 \text{ mm}$, [1,2]:

$$t_2 = \frac{R^2 - 2500}{35,332} = \frac{150^2 - 2500}{35,332} \approx 566 \text{ sec} \sim 9,43 \text{ min}$$

Promena ugaone brzine (ω_2) dobijamo zamenom brojnih vrednosti u izrazu (20) tj. njena vremenska promena, izraz (45), [2]:

$$\omega_2 = \frac{v}{\sqrt{r^2 + \frac{\delta}{\pi} \cdot v \cdot t}} = \omega_2 = \frac{150}{\sqrt{50^2 + \frac{0,74}{\pi} \cdot 150 \cdot t}}$$

$$\omega_2 = \frac{1}{\sqrt{0,111 + 0,0016 \cdot t}} = (0,111 + 0,0016 \cdot t)^{-0,5} \quad (45)$$

Promena momenta inercije namotavanja valjak i MAT-a, izraz (32) dobija se modifikovana vrednost zamenom (42 i 43) kao i diferenciranjem izraza (45) moment inercije se izražava izrazom (46) [2]:

$$M_{i2} = \frac{m_2 \cdot r_2^2}{2} \cdot \frac{d\omega_2}{dt} = \frac{30 + 0,169 \cdot t}{2} \cdot (2,5 + 35,332 \cdot t) \cdot \left(-\frac{1}{2}\right) \cdot (0,111 + 0,0016 \cdot t)^{-\frac{3}{2}} \cdot 0,0016 \quad (46.1)$$

$$M_{i2} = \frac{1}{(\sqrt{(0,111 + 0,0016 \cdot t)})^3} \cdot (15 + 0,08435 \cdot t) \cdot (-1) \cdot 0,008 \cdot (46.2)$$

$$M_{i2} = (15 + 0,08435 \cdot t) \cdot (2,5 + 35,332 \cdot 10^{-3} \cdot t) \cdot (-0,008) \cdot \omega_2^3 \quad (46.3)$$

Zamenom ukupnog vremena namotavanja tj. za $t = 566 \text{ [s]}$ i zamenom u izraz (42) dobija se ukupna težina namotanog valjka:

$$m|_{t_2=566} = 30 + 0,169 \cdot 566 = 30 + 95,65 = 125,65 \text{ kg}$$

Kako se namotavaju po dve rolne širine 1250 mm to se može izračunati težina jedne rolne:

$$(125,65 - 30) = \frac{95,65}{2} = 47,825 \text{ kg, što se može reći}$$

da odgovara i izmerenim težinama datim u tabeli T – 2 [2,8,14,15,16,17].

Sl.12 Strukturalna simulaciona šema mehanizma za "dotezanje" međuslojeva staklenih vlakana MAT-a proklizavanjem

Simulacijom izraza (40) prema simulacionoj šemi u MATLAB-u (sl. 12) dobijene su vremenske promene date na sl. 12 do 14 [2].

Sl. 13 Promena ugaone brzine namotača u funkciji prečnika namotavanja (vremena)

Sl. 13 Promena vučnog momenta uređaja za namotavanje u funkciji vremena

Sl. 14 Promena ugaone brzine rotora motora sa rastom prečnika namotavanja

Kao prilog sprovedene simulacione analize može da poslužiti sl.15 i tabela T -1 [2,8,15,19,20]. Ona pokazuju izgled namotane rolne a u tabeli su prikazani rezultati 18 merenja metodom slučajnog uzorka (pre ugradnje projektovanog uređaja).

Sl. 15 Izgled namotanih rolni bez ugrađenog uređaja za namotavanje sa zatezanjem »proklizavanjem« međuslojeva

Tabela T - 1

Redni broj	Izmerene težine namotanih rolni	Ssrednje vrednosti	Frekvencije pojavljivanja
1	28,50 – 29,50	29,00	6
2	29,50 – 30,50	30,00	5
3	30,50 - 31,50	31,00	4
4	31,50 – 32,50	32,00	3
UKUPNO			18

Na sl. 16 pokazan je izgled namotanih rolni MAT-a nakon što je ugrađen projektovani uređaja a što potvrđuje povećanje u izmerenoj težini. Podaci dobijeni slučajnim merenjem serije namotavanja od 18 rolni dati su u tabeli T – 2 [2,8,15,19,20].

Sl. 16 Izgled namotanih rolni nakon ugrade uređaja za namotavanje sa zatezanjem »proklizavanjem« međuslojeva

Tabela T - 2

Redni broj	Izmerene težine namotanih rolni	Ssrednje vrednosti	Frekvencije pojavljivanja
1	43,50 – 44,50	44,00	1
2	44,50 – 45,50	45,00	7
3	45,50 - 46,50	46,00	6
4	46,50 – 47,50	47,00	4
UKUPNO			18

Na slikama 17 i 18 prikazan je izgled izvedenog (elektro – mehaničko – pneumatsko) rešenje sistema za regulaciju promenljivih pogonskih parametara.

Sl. 17 Izgled mehanizma (radni i upravljački) za zatezanje proklizavanjem unutrašnjih međuslojeva (tvrdo namotavanje - radni deo)

Sl. 18 Izgled mehanizma (radni i upravljački) za zatezanje proklizavanjem unutrašnjih međuslojeva (tvrdo namotavanje - upravljački deo)

6. ZAKLJUČAK

Svrha ulaganja u revitalizaciju postojećih kao i uvođenje novih tehnologija je u smislu zadovoljenja: fleksibilnosti, adaptivnosti, otvorenosti a sistemi moraju pre svega biti efikasni. Prezentovani rezultati su pravi primer ovakvih sistema [2].

Regulacije promenljivih kinematskih dinamičkih pogonskih parametara (za vučeni valjak), datih izrazima (25 do 29), simuliranje na računaru izraz (40) a shodno strukturnim dijagramima sl. 11 i 12. Promena tekućih promenljivih parametara u radu je prezentovana za gramaturu 450 gr/m^2 i data je slikama 12 do 14. Sve veličine su dobijene u funkciji vremena t (izraz 32). Sa dijagrama, sl. 12, može se videti da se sa povećanjem vremena (prečnik namotavanja) ugaona brzina uređaja za namotavanje menja po eksponencijalnom zakonu, i to u odnosu ugaonih brzina. Na početku i na kraju namotavanja taj odnos je 1:3, tj. kakav je i odnos prečnika namotavanja. Slična karakteristika je dobijena sl. 14, za promenu ugaone brzine motora. Što se tiče promene momenta inercije namotavanog materijala sa sl. 13, karakteristika je skoro linearna, osim na samom početku namotavanja. Sa smanjenjem gramature raste ugaona brzina motora i uređaja za namotavanje (zbog debljine i veće dužine MAT-a) a moment inercije ima manju početnu vrednost, a samim tim i težinu namotanih rolni, [2].

Kao prilog ovih tvrdnji mogu nam poslužiti i tabele T – 1 (bez ugrađenog uređaja) i T – 2 (nakon ugradnje uređaja. Ista se odnosi na povećanje u težini namotavanja kao i boljeg sлагanja namotanog materijala. To znači da je namotavanje „tvrđe“, što i pokazuje sl. 16.

7. LITERATURA

- [1] Industrija staklenih vlakana–Boljevac: Tehnološka i konstrukcijska dokumentacija postrojenja za proizvodnju staklenih vlakana 1990. god.
- [2] Živković M.: Optimizacijamehaničkihparametara elektromehaničkogpogonasistemaza namotavanjestaklenihvlakana (MAT-a), mag. rad Tehnički fakultet u Čačku, 2000. god.
- [3] M. STAŃCZYK, T. FIGLUS (2016): A METHOD FOR THE SELECTION OF CERTAIN FORCE AND ENERGY PARAMETERS OF AUTOMATIC SHEET METAL COILING MACHINES, UDC – UDK 621.41.1:539.416:669.5:531.78:6=111, ISSN 0543-5846 METABK 55 (1), 79-82
- [4] Jeff Damour: THE MECHANICS OF TENSION CONTROL, CONVERTER ACCESSORY COPORATION Wind Gap, PA USA, www.converteraccessory.com/papers/tcpaper1
- [5] Hyeunhun Shin, Thanhtham Ho and Sangyoon L. (2010): Steering guide-based lateral control for roll-to-roll printed electronics, DOI 10.1007/s12206-009-1150-5, Journal of Mechanical Science and Technology 24 (319 – 322)
- [6] T. Ho, H. Shin and S. Lee, Fuzzy Control of the Lateral Position of a Moving Web in Roll-to-Roll Processes, Proc. of IEEE ROBIO, Bangkok, Thailand (2008) 1637-1642.
- [7] Steering guide system, http://www.fife.com/fife_products_fife_kantiroller.asp
- [8] Živković M., Golubović D.: Uperedna analiza provere saglasnosti normalnog zakona raspodele pri kontroli odstupanja izmerenih težina namotanih rolni asureod sečenih vlakana (MAT-a), 3. DQM konferencija „UPRAVLJANJE ODRŽAVANJEM“ V. Banja–Srbija, 3–4. februar 2000 (str. 215-225)
- [9] Miltenović V.: Mašinski elementi-oblici, proračun, primena, MF – Niš, 1997
- [10] Miltenović V.: Mašinski elementi-tablice i dijagrami, MF Niš, 1997.
- [11] Vučković V.: Električni pogoni, Elektrotehnički fakultet Beograd 1997. god.
- [12] Vukosavić D.: Controlledelectricaldrives-status of technology, XVI Konferencija za ETRAN, Vrnjačka Banja, 2. – 5. juna 1998. god.
- [13] VukosavićS.: Projektovanje adaptivnog mikroprocesorskog upravljanja brzinom i pozicijom asihronog motora, doktorska disertacija, Beograd 1989.
- [14] Živković M., Golubović D., Stojanović D., Antić B.: Prilog izbora načina upravljanja i vrste pogona uređaja ili mašine, V Međunarodni naučno–stručni skup – Izvor i prenos snage IPS 99, Podgorica–Bečići, 31. V - 3. VI 2000. CG–26599E11, (str. 37-44)
- [15] Živković M., Golubović D., Randić D.: Statističke metode i kvalitet proizvoda, NSČ–Menaddžment totalnim kvalitetom, No2 Vol 28, 2000, 27 godišnjica Jusk-a i III Međunarodni kongres Jusk-a (poslovna izvrsnost za trči milenijum, Beograd, 16–19. maj 2000 (str. 123-126)
- [16] WWu, D. H., Z. X. Tan, and H. Q. Yang." www.converteraccessory.com/papers/tcpaper1 *Pack King Engineering* 40.8 (2005):8-11.u,
- [17] YI HOU: NOVEL CONTROL APPROACHES FOR WEB TENSION REGULATION, Master of Science in Mechanical Engineering Beijing Institute of Technology, Beijing, P.R. China, 2001. (124)
- [18] Smolinski, P., Miller, C.S., Marangoni, R.D. & Onipede, D. (2002) Modeling the collapse of coiled material, *Finite Elements in Analysis and Design*, 38, 521-535,
- [19] Dašić P.: Statističke metode i sistem kvaliteta ISO 9000, Naučni skup SISTEMA KVALITETA uslov za uspešno poslovanje i konkurentnost Zbornik radova, Jugoslovensko udruženje za standardizaciju i kvalitet, podružnica Rasinskog okruga, Kruševac 25-26 novembar 1999, pp. 207 - 219.
- [20] Dašić, P.: Approximation of cutting tool wear function using polynomial regression equation. *Journal of Research and Development in Mechanical Industry (JRaDMI)*, Vol. 3, Issue 3 (September 2011), pp. 171-180. ISSN 1821-3103

The Specifics of the Process of Winding of the Stripped Material in the Form of Matting Made From Chopped Glass Fiber (MAT)

Dr Milutin Živković¹; MSc Jasmina Miljojković²; Ma Ivan Bijelić³

¹Technical College of Mechanical Engineering Professional Studies, 37240 Trstenik, Radoja Krstića 19, Serbia

²Faculty of Mechanical Engineering in Kragujevac, Sestre Janjić 6, 34000 Kragujevac, Serbia

³Technical College of Professional Studies, Bulevar Zorana Đinđića 152a, 11070 Novi Beograd, Serbia

The processes of winding (unwinding) are characterized by variable kinematic and dynamic parameters. Thus, in the case of increasing the diameter of the coiled material, its frequency of rotation is reduced, and vice versa. The implemented system is also specific in the manner of transmission of torque, using friction transmission. For these reasons, this paper mathematically defines the variable kinematic and dynamic drive parameters using the example of matting winding from staple glass fibers at ETEKS - from Baljevac. To make the process of winding to work through constant tractive force and linear speed, the power on the traction roller (winding) has to be constant. It must be such that it provides sliding between internal interlayers as well as avoiding the severance (tearing) of the base material. The obtained dependancies are simulated on a computer, for matting of 450g/m² grammage, and the results showed a significant agreement with the results of weight measurements of the wound-up mattings of the projected operating device.

Keywords:winding (unwinding), glass fibers, mathematical model, variable parameters, simulation, cardboard box, weight measurement of wound rolls.

0. INTRODUCTION

DP glass fiber industry "ETEX" Boljevac produced the alkali-free glass fibers (ROVING and MAT), which are of mineral origin, known under the international label "E", which implies the precise chemical composition of the glass (Fig.1) [1]. Installed annual capacity is about 5400 tons of finished products in the form of MAT and ROVING (ratio: 40% MAT, 60% ROVING). The company employed 400 workers and was built under the license of the English company "TBA - BISHOP" [1,2]. However, its fate is similar to the fate of many of our manufacturing companies, as it was closed at the end of 2010.

Fig.1A detail from the production of glass fiber

MAT was produced in the form of matting made of chopped glass fibers, which are bound together with a binding agent. It is produced as emulsional M1102 (internal mark) and powdered M1104, with a width ranging from 1000 to 3000 (mm), wound-up on cardboard rolls up to 110 (m) in length. After the winding, the same rolls are packed in cardboard boxes of dimensions 300 x 300 (mm), and are then palletized (18 boxes). The grammage of MAT ranges from 250 to 900

g/m² [1,2]. This paper explains the specificity of the transfer of torque as well as the method of winding at the output part of the production line (Fig. 2) [1,2]. It also explains all the specifics relating to the winding of these materials.

Fig.2The layout of the output section of the winding line

The type of the winding material (steel or copper strips, Alu-foils, textile materials or paper) influences the choice of drive modes, depending on both the mechanical and technological characteristics. From the experience of the author of this paper, as well as from the available literature data, it is known that for the winding (and unwinding) of such materials both electromotive and hydraulic drives are used [2,3]. In addition to the drive method, these systems must also take into account the curvature of the side edges as well as the possible reasons for damage done to the materials, i.e. edge regulation [2,4,5,6,7].

The basic requirement, which is required for all winding or unwinding devices is the constant linear speed, as well as the tension force in the material. The choice of the control device is carried out according to the criteria that should meet the requirements, taking into account the changing kinematic and dynamic parameters

of winding. As the problem of winding matting from chopped glass fiber is already a complex one, including the method of the transfer of torque (frictional transmission), this solution makes the above-mentioned problems even more complex. The obtained mathematical dependencies are examined on the simulation model and the results are supposed to be consistent with the actual weight measured on a device that has been redesigned [2,3,8].

1. TRANSMISSION OF TORQUE WHILE WINDING THE MATTING

Angular velocity and torque of the driving machine rarely coincide with the demands of work machine. The most commonly used asynchronous AC motors have 2, 4 and 6 poles, and at the frequency $f = 50\text{Hz}$ the number of engine revolutions are 3000, 1500 or 750 min^{-1} . Internal combustion engines are also working economically in a narrow field of frequency speed. Work machines usually require high torque at low speeds, so it is necessary to install the power transmitter between the driving and the work machine [2,9,10,11,12].

The task of the transmitter is the conversion and "conduction" of mechanical energy from the output shaft of the driving machine (motor, turbine) to where it is needed - the input shaft of the work machine, i.e. the driving wheels of the mobile instrument. Depending on the conditions, which are dictated by work machines, transmitters are either constant or of variable transmission ratio. In general it can be said that the transmitter unit consists of three parts - the input and output shaft, and the casing, in which the shafts are interconnected based on the mechanical, electrical hydraulic or pneumatic principles. If the connection between the drive and the driven shaft is based on a mechanical principle, then those are mechanical transmitters. They transmit energy i.e. torque and the angular velocity in two main ways: through shape and friction (adhesion). Transmission can be accomplished by the direct touch of the drive and the driven element, or through an intermediary.

Torque between the drive and the upper shaft can be transmitted by friction as well, i.e. indirect touching. When turning the friction wheel of the drive on the contact area, a friction force arises, leading to the rotation of the other - drive friction wheel. In order for the friction transmitter to work, it is necessary that the friction force (F_μ) is always greater or in the borderline case equal to the tangential, i.e. circumference force (F_t) on the friction wheels (equation (1)):

$$F_\mu \geq F_t \quad (1)$$

The required frictional force on the contact surfaces of friction wheels is obtained by the mutual pressure of one friction wheel on the other. All the advantages of these transmitter units are well known and covered in fine detail in the scientific literature [9,10]. The choice of materials for the friction pairs plays an important role in the capacity and lifetime of the transmitter. It must meet the following requirements:

- A high resistance in relation to the touch pressure, and with respect to wear, in order to provide the with service life of the transmitter unit (annealed steel);
- A large friction coefficient in order for the given

pressure force to provide a large friction force (rubber),

- Great modulus of elasticity to minimize the elastic deformations, and therefore the elastic sliding.

The shape of the working surfaces of friction pairs is closely linked to the relationship of angular velocity and the mutual position of the axes of rotation of the friction wheels. Specifically, it must meet the condition of rolling without sliding of one work surface over another, which means that circumferential speeds of connection points must be the same (Fig. 3) [2,9,10]. Values, which determine the position of one rotational axis related to the other, as well as the ratio of the intensity of the corresponding angular velocities are: axial distance and kinematic transmission ratio.

Fig.3 Winding up of frictional contact pairs for the transmission of torque (winding of MAT)

Axial spacing is the shortest distance of the axes of rotation (min and max) given by the equation (2): [2]

$$\frac{D_A + d}{2} = a_1 \leq a_2 \leq \frac{D_A + D}{2} = a_{\max} \quad (2)$$

Circumferential velocity (v) is given by equations (3 and 4) [2]:

$$v = \frac{D_A}{2} \cdot \omega_1 = \text{const} \quad (\text{for the friction wheel 1}) \quad (3)$$

$$v_1 = \frac{d_2}{2} \cdot \omega_2 = \frac{2 \cdot r_2}{2} \cdot \omega_2 = r_2 \cdot \omega_2 \quad (4)$$

One can calculate the nominal kinematic ratio (i_0), as the ratio of the intensity of the vectors of angular velocities. It is always a positive value, greater than or equal to 1 (where true equality of velocity applies, i.e. velocity $v_1 = v_2 = v = \text{const}$), seen in the equation (5) [2,9,10].

$$i_0 = \frac{\omega_1}{\omega_2} = \frac{\frac{d_2}{2} \cdot v_2}{\frac{D_A}{2} \cdot v_1} = \frac{d_2}{D_A} \quad (5)$$

Due to the elastic sliding between the contact surfaces of the friction pair, the actual transmission ratio (i) deviates from the normal, seen in the equation (6) [2,9,10]:

$$i = \frac{i_0}{1 - \epsilon} \quad (6)$$

In order to transfer the torque through friction transmission, the necessary and sufficient condition is given in the equation (1), and the transfer mechanism is shown in Figure 4.

In Figure 5, the forces for the drive wheel and the winding drum are split apart [2].

Fig. 4 The forces at the site of contact of the friction pair

On the basis of the equation (1) as well as the Figure 5, we get the equation (7), ie. the limiting case [2,9,10]:

$$F_N \cdot \mu = S_\mu \cdot F_t \quad (7)$$

Where: F_N [kN] - is the normal force (achieved on the trolley mechanism by using pneumatic cylinders, and depending on the air pressure in the pneumatic system) μ - is the coefficient of friction of glass fibers at the point of contact, S_μ - the degree of safety against sliding is 1,5... 2 (for the exact shape the axis without kinematical sliding) [2,9,10].

Fig. 5 Forces at the point of touch of the friction pair

F_t [kN] – the tangential force on the circumference of the drive wheel, which can be calculated using the equation (8) [2,9]:

$$F_t = \frac{T_1}{\frac{D_A}{2}} = \frac{2 \cdot T_1}{D_A} \quad (8)$$

In accordance with equation (8), for the transmission of the circumferential force a normal force between the contact surfaces is needed, and can be expressed with the equation (9):

$$F_N = \frac{S_\mu \cdot F_t}{\mu} = 2 \cdot \frac{T_1 \cdot S_\mu}{\mu \cdot D_A} \quad (9)$$

It is constant for the entire duration of the winding process.

2. MECHANICS OF WINDING OF FLEXIBLE MATERIALS

Many technological processes in the area of "finalising" (fabrics and paper) as well as thickness reduction (foils, steel sheets and sheets of non-ferrous materials) require fast rewinding of the material from one cylinder to another and vice versa. The process requires that the linear (transport) speed of the "finalized" material is constant for the whole time of the winding. The problem with winding consists in seeking the most accurate mathematical formulation and the establishment of quantitative and logical dependance of the operational parameters. In addition to traditional mechanical systems, hydrostatics has also found its place in solving these problems, by developing the components

that have enabled it to. In recent years, the standard electric drives are increasingly finding their place in solving these problems through the development of the elements of the control electronics [2,3,4,5,6,7,11,12,]. To solve these problems, regardless of the type of drive, it is necessary to first set all of the geometric, kinematic and dynamic functional parameters.

2.1 System geometry

Based on the equality of volume of material, before and after winding on the drums (rollers), in accordance with Fig. 6, a dependence can be established with the equation (10) [2,3,7]:

$$L \cdot \delta \cdot b = (D^2 - d^2) \cdot \frac{\pi}{4} \cdot b \quad (10)$$

Where: L [m] – is the length of the winding material, δ [m] – is the thickness of the winding material, b [m] – is the width of the winding material, $D = 2R$ [m] – is the diameter wound roller, $d = 2r$ [m] – the outer diameter of the empty (non-wound) roller.

Fig. 6 General case of the rewinding of the strip material from one roller to another

On the basis of the equation (10), the length of the coiled material, as a function of the radius of the wound-up roller and the thickness of the winding material, is given by the equation (11) [2]:

$$L = (R^2 - r^2) \cdot \pi / \delta \quad (11)$$

The equation (11) may be illustrated by the diagram shown in the Fig. 7 [2].

Fig. 7 The dependence of the thickness and length a wound material to the constant diameter of the winding

If we consider the general case, in that the part of the material rewound from the roller 1 to the roller 2 as seen on Fig. 6, from the conditions of equality of length we get the equation (12):

$$L = L_1 + L_2 \quad (12)$$

Where: L_1 [m] – length of the material on roller 1,
 L_2 [m] – length of the material on roller 2.

Based on the terms given with the equation (12) we can get the equation (13):

$$(R^2 - r^2) \cdot \frac{\pi}{\delta} = (r_1^2 - r^2) \cdot \frac{\pi}{\delta} + (r_2^2 - r^2) \cdot \frac{\pi}{\delta} \quad (13)$$

After tying up, we get the equation (14):

$$r_1^2 + r_2^2 = R^2 + r^2 = \text{const} \quad (14)$$

From the equation (14) it can be concluded that the dependance of ongoing parameters changes according to the equation of the circle, which can be illustrated by Fig. 8 [2].

Fig. 8 Changes in "ongoing" diameters of winding (unwinding)

2.2 The kinematics of the system

Determination of the kinematic quantities will be observed for both the "tow roller" (roller on which the desired material is wound on) and for the "towed roller" (roller from which material is unwound). According to Fig. 6 the tow roller is marked with the number 2, and the towed roller with number 1 [2].

Tow roller: From the conditions of equality of linear speed of winding materials ($v = \frac{dL_2}{dt} = \text{const}$), length L_2 can be determined in accordance with the equation (15):

$$L_2 = v \cdot t \quad (15)$$

Where: v [m/sec] – is linear speed (circumferential velocity of the tow roller) and t [s] – is time.

Substituting equation (15) for (11) gives us the equation (16):

$$v \cdot t = (r_2^2 - r^2) \cdot \frac{\pi}{\delta} \quad (16)$$

According to the equation (16) a variable parameter (r_2) can be expressed by the equation (17):

$$[2]r_2 = \sqrt{r^2 + \frac{\pi}{\delta} \cdot v \cdot t} \quad (17)$$

The equation (17) shows the change in the radius of the tow roller as a function of time, but seeing that the circumferential velocity is constant, her change will be expressed by (18):

$$\omega_2 = \frac{v}{r_2} \quad (18)$$

Where: ω_2 [s⁻¹] – is the angular velocity of the tow roller

When the change in angular speed of the tow roller (drum) is known, we can determine the number of revolutions of the tow roller, based on well-known term in mechanics for angular velocity ($\omega = \frac{n \cdot \pi}{30}$).

Number of revolutions of the towed roller (n_2) can be calculated in accordance with the equation (19):

$$n_2 = \frac{30 \cdot v}{\pi \cdot r_2} \quad (19)$$

Linear speed is constant (project requirement) and one for the towed drum can be determined according to the equation (20):

$$v = r_2 \cdot \omega_2 = \text{const} \quad (20)$$

Differentiating the equation (20) by time gets us the term given by the equation (21):

$$\frac{dv}{dt} = \frac{dr_2}{dt} \cdot \omega_2 + r_2 \cdot \frac{d\omega_2}{dt} = 0 \quad (21)$$

Seing as $\frac{dr_2}{dt} = \frac{\delta \cdot v}{2\pi \cdot r_2}$ we get the equation (22)[2]:

$$\frac{d\omega_2}{dt} = \dot{\omega}_2 = \frac{\delta \cdot v^2}{2\pi \cdot r_2^3} \quad (22)$$

Where: $\dot{\omega}_2$ [sec⁻²] - angular acceleration of the tow roller.

Towed roller: The remaining amount of material on the towed roller, in the observed moment, may be determined in accordance with the equations (23 and 24)

$$L_1 = L - L_2 \quad (23)$$

$$(r_1^2 - r^2) \cdot \frac{\pi}{\delta} = (R^2 - r^2) \cdot \frac{\pi}{\delta} - v \cdot t \quad (24)$$

By solving the equation (24) at unknown (r_1) the equation in the equation (25) is obtained:

$$r_1 = \sqrt{R^2 - \frac{\delta}{\pi} \cdot v \cdot t} \quad (25)$$

Equation (25) shows the change in the radius of the towed roller in a function of time. If you repeat the entire course of calculation for the towing roller we get the following equations (26, 27 and 29) [2]:

Angular velocity of the towed roller:

$$\omega_1 = \frac{v}{r_1} [\text{s}^{-1}] \quad (26)$$

Number of revolutions of the towed roller:

$$n_1 = \frac{30 \cdot v}{\pi \cdot r_1} [\text{min}^{-1}] \quad (27)$$

$$\text{Angular acceleration: } \dot{\omega}_1 = \frac{d\omega_1}{dt} = \frac{\delta \cdot v^2}{2\pi \cdot r_1^3} [\text{s}^{-2}] \quad (28)$$

Since the torque transmission and the drive of the "towing roller" (on which the strips of glass fiber are wound) is achieved through the friction transfer, all of the equations for the towing roller apply, which can be illustrated by Fig. 9 [2].

$$\omega_2 = \frac{v}{r_2} \quad (29)$$

On the basis of the equation (18), wherein the linear speed during the entire time of winding ($v = \text{const}$), we get the equation (29) that gives the dependence of the angular speed (number of revolutions) of the winding material to the diameter of winding (r_2) [2].

Fig. 9 The kinematics of the winding of glass fibers (MAT)

2.3 Dynamics of the winding system

The dynamics of the drive of the device for winding stripped materials falls into the domain of issues of mechanisms with variable mass. Designed winding mechanism uses sliding interlayers, which makes it possible to eliminate the appearance of loose winding, as shown schematically in Fig. 10 [2].

Based on this, a winding mechanism with the sliding of the internal interlayers was projected i.e. the "loose" built-in winding was eliminated [1,2].

As the thickness of the strips of the stripped material of glass fibers of MAT is small compared to the initial diameter of the unwound cardboard tube ($\phi d \gg \delta$) it can be considered that the brunt axis of the winder matches the axis of rotation of the device without the tape.

Fig. 10 Kinematic scheme of the mechanism for tightening of internal interlayers through sliding (Hard-winding)

The mass of the roller (m) in an arbitrary point of winding is given by the equations (30 and 30a) [2]:

$$m = m_0 + m_2 \quad (30)$$

$$m = m_0 + L_2 \cdot b \cdot \delta \cdot \rho \quad (30a)$$

Where: $L_2[m]$ – the length of a wound material at some point in time

$m_2[kg]$ – mass of the wound material as a function of the winding radius r_2 (a function of time t),

$m_0=30[kg]$ – mass of the empty roller without the wound materials,

$\delta[m]$ – the thickness of the wound material,

$\rho[kg/m^3]$ – density of the material,

$b[m]$ – width of the wound material.

Constructively it may be set that the mass of the empty roller has a value ($m_0 = \pi \cdot b \cdot \rho \cdot r^2$) so identically we can write an equation (31):

$$m_2 = \pi \cdot b \cdot \rho \cdot r_2^2 \quad (31)$$

Moment that comes from the inertia of the roller and the wound material is given by equation (32):

$$M_{i2} = J_2 \cdot \frac{d\omega_2}{dt} = \frac{m_2 \cdot r_2^2}{2} \cdot \frac{d\omega_2}{dt} \quad (32)$$

Based on equations (3) and (22) and the switch in (32) we get the equation (33.1 i 33.2) [2,12,13,14]:

$$M_{i2} = \frac{\pi \cdot b \cdot \rho \cdot D_2^2}{2} \cdot r_2^2 \cdot \frac{d\omega_2}{dt} \quad (33.1)$$

$$M_{i2} = \frac{\pi \cdot b \cdot \rho}{2} \cdot \left(\frac{v}{\omega_2}\right)^4 \cdot \frac{d\omega_2}{dt} \quad (33.2)$$

For the angular velocity of the wound roller we can write an equation (34):

$$\omega_2 = \frac{\omega_m}{i} = \frac{\omega_m}{i_f \cdot u_1} \quad (34)$$

Where the equivalent transmission ratio of rollers is:

$$i = i_f \cdot u_1 = 47.84 \cdot 0.912 = 43.63 \quad (35)$$

On the basis of the equations (32), (33), (34) and (35) we can use the equation (36) to establish a link between the moment of winding and the moment on the drive axis of motor shaft, using the equations (36.1, 36.2 and 36.3) [2,12,13, 14,16,17,18]

$$M_{ieM} = \frac{M_{i2}}{i} = \frac{\pi \cdot b \cdot \rho \cdot v^4 \cdot i^4}{2} \cdot \frac{1}{\omega_m^4} \cdot \frac{d\omega_m}{dt} \quad (36.1)$$

$$M_{i2} = \frac{\pi \cdot b \cdot \rho \cdot v^4 \cdot i^3}{2} \cdot \frac{1}{i} \cdot \frac{d\omega_m}{dt} \quad (36.2)$$

$$M_{i2} = \frac{\pi \cdot b \cdot \rho \cdot v^4 \cdot i^2}{2} \cdot \frac{d\omega_m}{dt} \quad (36.3)$$

On the other hand, the equation of dynamic equilibrium reduced to the shaft of the drive electromotor can be expressed by using the equation (37) [2]:

$$M_e = J_m \cdot \dot{\omega}_m + K_t \cdot \omega_m + M_{ieM} \quad (37)$$

Where: $J_m[kg \cdot m^2]$ – is the moment of inertia of an electromotor shaft and all its working parts,

K_t – friction coefficient,

M_e – electromagnetic moment,

M_{ieM} – moment reduced on the motor shaft.

Based on equations (36.1) i (37) we can get the equation (38) [2,6,7,12,13]:

$$M_{ieM} = \left(J_m + K_{Me} \cdot \frac{1}{\omega_m^4} \right) \cdot \frac{d\omega_m}{dt} + K_t \cdot \omega_m \quad (38)$$

For the electric part of the electromotor as well as the voltage inverter with power regulation, we get the equation (39) [2,12,13]:

$$M_e = K_{inv} \cdot \omega_{mc} \quad (39)$$

Where: K_{inv} – inverter constant (together with the electric part of the electromotor),

ω_{mc} – reference speed of the electromotor.

In accordance with the equality of electromagnetic moments, given in the equations (38) and (39) we obtained the final configuration of the equation for the simulation (40):

$$\left(J_m + K_{Me} \cdot \frac{1}{\omega_m^4} \right) \cdot \frac{d\omega_m}{dt} + K_t \cdot \omega_m = K_{inv} \cdot \omega_{mc} \quad (40)$$

The resulting equation can be represented using a structural block diagram shown in Fig. 6 [2,12,13].

Fig. 11 Block diagram of the winding device with sliding interlayers (4.22),

In order for circumferential velocity to be constant, the necessary requirement is that the reference angle (ω_{mc}) of the velocity of the motor (in time) should be changed according to the law given by the equation (41) [2,12,13]:

$$\omega_{mc} = \frac{v \cdot t}{\sqrt{r^2 + \frac{\rho \cdot v}{\pi} \cdot t}} \quad (41)$$

3.SIMULATION RESULTS

Simulating the equation (40), presented in the form of a structural block diagram in Fig. 11, the regulation parameters were simulated on a computer, using the "MATLAB" programming language. As an aid, the simulation schemes on Fig.12 were used, as well as the kinematic scheme from Fig. 10 [2]. Inspection of the kinematic and dynamic parameters was carried for MAT, the grammages in ETEx are, from 450 [g/ m²] and the auxiliary data for the calculation are as follows: [2]:

$$r = \frac{d}{2} = \frac{100}{2} = 50 \text{ [mm]} - \text{the radius of the non-wound cardboard tube;}$$

$$R = \frac{D}{2} = \frac{300}{2} = 150 \text{ [mm]} - \text{the radius of the wound material;}$$

$$L = 85 \cdot 10^3 \text{ [mm]} - \text{the specified optimal length of the MAT tape;}$$

$$b = 2500 \text{ [mm]} - \text{width of the winding tape (2x1250m);}$$

$$v \sim 9 \text{ [m/min]} - \text{linear speed of the tape at the exit from the dryer, i.e. at the winding device;}$$

$$m_0 = 30 \text{ [kg]} - \text{mass of the rotating parts of the tape winder.}$$

The change of the angular velocity (ω_m), at the electromotor axis (reductor input), and according to the characteristics of the embedded system is shown on the kinematic scheme in Fig. 10. Electric moment of the motor (either electromagnetic or the moment of conversion) is obtained when the mechanical output of the engine is divided by the angular speed of the rotor ω_m , as indicated by the calculated values [2]

$$\omega_m = \frac{\pi \cdot n_m}{30} = \frac{\pi \cdot 1380}{30} = 144,5 \text{ [s}^{-1}\text{]}$$

$$T_m = \frac{P_m}{\omega_m} = \frac{0,55 \cdot 10^3}{144,5} = 3,81 \text{ [N m]}$$

$$T_{\text{numax}} = T_m \cdot i_r \cdot u_l$$

$$T_{\text{numax}} = 3,81 \cdot 47,84 \cdot 0,912 \approx 166,2 \text{ Nm}$$

The total area of the strip:

$$L \cdot b = 85 \text{ [m]} \cdot 2,5 \text{ [m]} = 212,5 \text{ m}^2$$

- The total weight of the rolled strip:

$$m = \rho_p \cdot L \cdot b = 0,450 \text{ [kg} \cdot \text{m}^{-2}\text{]} \cdot 212,5 \text{ [m}^2\text{]}$$

$$m = 95,625 \text{ [kg]}$$

- The volume of the rolled strip:

$$V = L \cdot b \cdot \rho = 85 \text{ [m]} \cdot 2,5 \text{ [m]} \cdot 0,74 \cdot 10^{-3} \text{ [m]}$$

$$V = 0,157 \text{ [m}^3\text{]}$$

- The thickness of the strip will be achieved with the equation(2):

$$\delta = \frac{(R^2 - r^2)}{L} \cdot \pi = \frac{(150^2 - 50^2)}{85 \cdot 10^3} \cdot \pi = 0,74 \cdot 10^{-3} \text{ m}$$

- Volume density:

$$\rho = \frac{m}{V} = \frac{\rho_p \cdot L \cdot b}{L \cdot b \cdot \delta} = \frac{\rho_p}{\delta} = \frac{0,45}{0,74 \cdot 10^{-3}} = 608,11 \text{ kg/m}^3$$

According to the equation (2) obtained by replacing the changes (increase) in mass in a function of time (t), the equation 21a can be expressed by the new equation (42) [2]:

$$m = 30 + 0,15 \cdot 2,5 \cdot 0,74 \cdot 10^{-3} \cdot 608,11 \cdot t$$

$$m = 30 + 0,169 \cdot t \text{ [kg]} \quad (42)$$

The change of winding diameter, according to the equation (8), where through the replacement of the calculated values gets us its time change in the equation (43):

$$r_2 = \sqrt{r^2 + \frac{\delta}{\pi} \cdot v \cdot t} = \sqrt{50^2 + \frac{0,74}{\pi} \cdot 150 \cdot t}$$

$$r_2 = \sqrt{2500 + 35,332 \cdot t} \text{ [mm]} \quad (43)$$

Using the equation (43) one can calculate the total winding time, the equation (44):

$$t = \frac{r_2^2 - 2500}{35,332} \quad (44)$$

In the case of the end of winding ($r_2 = R$) we can determine, in accordance with the equation (36), the total time of winding (t_2). The maximum diameter of winding must in any case satisfy the condition of their placement in the boxes of standard dimensions 300 × 300 mm, [1,2]:

$$t_2 = \frac{R^2 - 2500}{35,332} = \frac{150^2 - 2500}{35,332} \approx 566 \text{ sec} \sim 9,43 \text{ min}$$

The change of the angular velocity (ω_2) obtained by replacing the numerical values in the equation (20), ie. its time change is given in the equation (45): [2]

$$\omega_2 = \frac{v}{\sqrt{r^2 + \frac{\delta}{\pi} \cdot v \cdot t}} = \omega_2 = \frac{150}{\sqrt{50^2 + \frac{0,74}{\pi} \cdot 150 \cdot t}}$$

$$\omega_2 = \frac{1}{\sqrt{0,111 + 0,0016 \cdot t}} = (0,111 + 0,0016 \cdot t)^{-0,5} \quad (45)$$

The change of the moment of inertia of winding of the roller and the MAT, equation (32) gives us the modified value through replacement (42 i43) as well as the differentiation of the equations (45). The moment of inertia is expressed by the equation (46) [2]:

$$M_{i2} = \frac{m_2 \cdot r_2^2}{2} \cdot \frac{d\omega_2}{dt} = \frac{30 + 0,169 \cdot t}{2} \cdot (2,5 + 35,332 \cdot t) \cdot \left(-\frac{1}{2}\right) \cdot (0,111 + 0,0016 \cdot t)^{-\frac{3}{2}} \cdot 0,016 \quad (46.1)$$

$$M_{i2} = \frac{(15 + 0,08435 \cdot t) \cdot (-1) \cdot 0,008}{(\sqrt{0,111 + 0,0016 \cdot t})^3} \quad (46.2)$$

$$M_{i2} = (15 + 0,08435 \cdot t) \cdot (2,5 + 35,332 \cdot 10^{-3} \cdot t) \cdot (-0,008) \cdot \omega_2^3 \quad (46.3)$$

Substituting the total time of winding for $t = 566 \text{ [s]}$ and substituting it in the equation (42) results in an overall weight the wounded roller:

$$m|_{t=566} = 30 + 0,169 \cdot 566 = 30 + 95,65 = 125,65 \text{ kg}$$

Seeing that the two rolls of the 1250mm width are used for winding, we can calculate the weight of one of the rolls:

$$(125,65 - 30) = \frac{95,65}{2} = 47,825 \text{ kg, which can be said}$$

that it corresponds with the measured weights given in the table T – 2 [2,8,14,15,16,17].

Fig.12 Structural simulation scheme of the mechanism for "tightening" intermediate layers of glass fibers of MAT through sliding/

By simulating the equation (40) according to simulational patterns in MATLAB (Fig. 12) the time changes on figures 12 to 14 were achieved [2].

Fig. 13 The change in the angular velocity winder in function of diameter winding (time)

Fig. 13 Changing the traction moment of winding device in a function of time

Fig. 14 The change of the angular velocity of the motor rotor with the increase in diameter winding

Contribution implemented as simulation analysis can serve Fig.15 and Table T -1 [2,8,15,19,20]. It shows a set of the wound roll and in the table shows the results of 18 measurements of a random sample (before the application of the designed device).

Fig. 15 The appearance of wound rolls without a built-in device for winding with the tensioning sliding of the interlayers

Table T - 1

Serial number	The measured weights of coiled rolls	Average values	Frequency of occurrence
1	28,50 – 29,50	29,00	6
2	29,50 – 30,50	30,00	5
3	30,50 – 31,50	31,00	4
4	31,50 – 32,50	32,00	3
TOTAL			18

In Fig. 16 is shown the layout of coiled rolls of MAT after embedded device designed which confirms an increase in the measured weight. The data obtained by measurement of a random series of winding rolls 18 are provided in a table T - 2 [2,8,15,19,20].

Fig. 16 The appearance of wound rolls after the installation of the winding device with the tightening "sliding" of the interlayers

Table T - 2

Serial number	The measured weights of coiled rolls	Average values	Frequency of occurrence
1	43,50 – 44,50	44,00	1
2	44,50 – 45,50	45,00	7
3	45,50 – 46,50	46,00	6
4	46,50 – 47,50	47,00	4
TOTAL			18

In figures 17 and 18 the appearance of the constructed (electro - mechanical - pneumatic) system solution for regulating variables operational parameters is shown.

Fig. 17 The appearance of the mechanism (operating and control) for tightening through sliding the internal interlayers (solid winding - working part)

Fig. 18 The appearance of the mechanism (operating and control) for tightening through sliding of the internal interlayers (hard winding - the control section)

4. CONCLUSION

The purpose of investing in the revitalization of existing and implementation of new technologies is done to meet the following: flexibility, adaptability, openness and most of all, the efficiency of the systems. The results presented here are the true example of such systems [2].

Regulation of variable kinematic and dynamic drive parameters (for the track roller), given with equations (25 to 29), and the computer simulation (equation 40) is done according to the structural diagrams in Fig. 11 and Fig. 12. Changing the flowing variables during work is presented for the grammage of 450 g/m², and is given in Figures 12 through 14. All values are obtained as a function of time t (equation 32). From the diagram (Fig. 12), it can be seen that with the constant build-up of time t (winding diameter) angular velocity of the winding machine changes through the exponential law, relative to angular velocities. At the beginning and the end of winding this ratio is 1: 3, ie. as is the ratio of the diameter winding. A similar characteristic is obtained (Fig. 14) for changing the angular velocity of the motor. Regarding the change of the moment of inertia of the winder material (Fig. 13) the characteristic is almost linear, except at the very beginning of winding. With a decrease in grammage angular speed of the motor and the winding device grows (due to the thickness and a greater length of MAT), the moment of inertia has a lower initial value, and therefore the weight of coiled rolls as well [2]. As an addition to these claims we may also use tables T-1 (without the built-in device), and T-2 (after the installation of the device). The same applies to an increase in the weight of winding as well as the better stacking of the wound material. This means that the winding is "harder", which Fig. 16 shows.

7. LITERATURE

- [1] Industrija staklenih vlakana–Boljevac: Tehnološka i konstrukcijska dokumentacija postrojenja za proizvodnju staklenih vlakana 1990. god.
- [2] Živković M.: Optimizacija mehaničkih parametara elektro mehaničkog pogona sistema za namotavanje staklenih vlakana (MAT-a), mag. rad Tehnički fakultet u Čačku, 2000. god.
- [3] M. STAŃCZYK, T. FIGLUS (2016): A METHOD FOR THE SELECTION OF CERTAIN FORCE AND ENERGY PARAMETERS OF AUTOMATIC SHEET METAL COILING MACHINES, UDC – UDK 621.41.1:539.416:669.5:531.78:6=111, ISSN 0543-5846 METABK 55 (1), 79-82
- [4] Jeff Damour: THE MECHANICS OF TENSION CONTROL, CONVERTER ACCESSORY COPORATION Wind Gap, PA USA, www.converteraccessory.com/papers/tcpaper1
- [5] Hyeunhun Shin, Thanhtram Ho and Sangyoon L. (2010): Steering guide-based lateral control for roll-to-roll printed electronics, DOI 10.1007/s12206-009-1150-5, Journal of Mechanical Science and Technology 24 (319 – 322)
- [6] T. Ho, H. Shin and S. Lee, Fuzzy Control of the Lateral Position of a Moving Web in Roll-to-Roll Processes, Proc. of IEEE ROBIO, Bangkok, Thailand (2008) 1637-1642.
- [7] Steering guide system, http://www.fife.com/fife_products_fife_kantiroller.asp
- [8] Živković M., Golubović D.: Uporedna analiza provere saglasnosti normalnog zakona raspodele pri kontroli odstupanja izmerenih težina namotanih rolni asureod sečenih vlakana (MAT-a), 3. DQM konferencija „UPRAVLJANJE ODRŽAVANJEM“ V. Banja–Srbija, 3–4. februar 2000., (str. 215 - 225)
- [9] Miltenović V.: Mašinski elementi-oblici, proračun, primena, MF – Niš, 1997
- [10] Miltenović V.: Mašinski elementi-tablice i dijagrami, MF Niš, 1997.
- [11] Vučković V.: Električni pogoni, Elektrotehnički fakultet Beograd 1997. god.
- [12] Vukosavić D.: Controlled electrical drives-status of technology, XVI Konferencija za ETRAN, Vrnjačka Banja, 2. – 5. juna 1998. god.
- [13] Vukosavić S.: Projektovanje adaptivnog mikro - procesorskog upravljanja brzinom i pozicijom asihronog motora, doktorska disertacija, Beograd 1989.
- [14] Živković M., Golubović D., Stojanović D.: Prilog izbora načina upravljanja i vrste pogona uređaja ili mašine, V Međunarodni naučno–stručni skup – Izvor i prenos snage IPS 99, Podgorica–Bečići, 31. V - 3. VI 2000. CG–26599E11, (str. 37-44)
- [15] Živković M., Golubović D., Randić D.: Statističke metode i kvalitet proizvoda, NSS–Menadžment totalnim kvalitetom, No2 Vol 28, 2000, 27 godišnjica JUSK-a i III Međunarodni kongres JUSK-a (poslovna izvrsnost za trči milenijum, Beograd, 16–19. maj 2000 (str. 123-126)
- [16] WWu, D. H., Z. X. Tan, and H. Q. Yang, "www.converteraccessory.com/papers/tcpaper1 Packing Engineering 40.8 (2005):8-11.u,
- [17] YI HOU: NOVEL CONTROL APPROACHES FOR WEB TENSION REGULATION, Master of Science in Mechanical Engineering Beijing Institute of Technology, Beijing, P.R. China, 2001. (124)
- [18] Smolinski, P., Miller, C.S., Marangoni, R.D. & Onipede, D. (2002) Modeling the collapse of coiled material, Finite Elements in Analysis and Design, 38, 521-535,
- [19] Dašić P.: Statističke metode i sistem kvaliteta ISO 9000, Naučni skup SISTEMA KVALITETA uslov za uspešno poslovanje i konkurentnost Zbornik radova, Jugoslovensko udruženje za standardizaciju i kvalitet, podružnica Rasinskog okruga, Kruševac 25-26 novembar 1999, str 207 - 219.
- [20] Dašić, P.: Approximation of cutting tool wear function using polynomial regression equation. *Journal of Research and Development in Mechanical Industry (JRadMI)*, Vol. 3, Issue 3 (September 2011), pp. 171-180. ISSN 1821-3103.