Energy characteristics of the tensioners motor when winding products of a wet composite material

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Abstract. Products made of composite materials by wet winding are very prevalent in different industries. Herewith, the product range is expanding and the quality requirements are constantly growing. The qualitative characteristics of future products largely depend on the properties of a tension control system. In its turn, the right choice of a tensioner drive power determines the system operation. Up to now, the main selection criterion of a tensioner electric drive was the ensuring the specified maximum tension of the composite tape less regard to the specific features of the used tensioner and the geometry of a wound product. However, when winding a product of a complex geometric shape, the dynamic moments arise in the electric drive and cause additional engine heating. This paper deals with the selection of the tensioners drive power taking into account the peculiarities of the wet composite material products winding

1. Introduction

The products made by winding of a composite material have unique chemical and mechanical characteristics and are applied in many industries: in spacecraft and rockets, in aviation, in the chemical industry [1-6]. The dry and wet winding are widely used.

Figure 1 shows the simplified scheme of a wet winding path.

![Figure 1. Wet winding of the product](image-url)
The bobbins with the material 2 is installed on the bobbin carrier 1. The number of the bobbins depends on the process operation and can vary from 6 to 1200 pieces. The dry material 3 passes through the tensioners 4 and is formed into the string on the special comb 5. The string passes through the bath 6 with the binder. A special sensor 7 controls the amount of binder. The squeezing rollers 8 remove the excess of binder. The electric drive 9 changes the gap between the rollers. The tensioner 10 consists of a fixed roller 11 and a roller 12 that moves along the tensioner by applying two parallel drives: an electric drive 13 and a pneumatic drive 14. By this way, the coverage angle of the rollers 11 and 12 by the wet tape changes. The tension of the tape itself changes as well. The measurer 15 measures the tape tension. The roller 16 lays the tape on the product 17 along a special path. For this purpose, it is equipped with stacking mechanisms, which is not shown on the figure. The drive 18 rotates the product 17. Several interconnected systems control the winding. The system 19 controls the dipping process, the systems 20 and 21 control the tension, the system 22 controls the rotation. The unit 22 controls all winding parameters and operates its process.

2. Problem formulation

The quality of the wound products is largely determined by the tension of the “wet” composite tape, with which it is wound on the mandrel [7-13]. The winding tension depends on the operation of the tensioner and the tension control system. However, to now, the main selection criterion of a tensioner electric drive was the ensuring the specified maximum tension of the composite tape less regard to the specific features of the used tensioner and the geometry of a wound product. However, when winding a product of a complex geometric shape, the dynamic moments arise in the electric drive and cause additional engine heating. Until now, this issue has not been investigated. Therefore, in practice, the developers were basing on their intuition when choosing the motor of the tensioner used when winding products with a “wet” composite tape.

There is little literature devoted to the choice of the power of the tensioners’ electric drive. We have managed to find only a few works on this issue [14-16]. However, in all works, there is no consideration for the specifics of winding composite products. For example, when winding a product with a complex geometric shape, the dynamic moments in the electric drive, which cause the additional heating of the motor, are not taken into account.

3. Theoretical part

When winding products with a wet composite tape, all the rollers of the winding path remain stationary. The tape slides along the path guides and the tension increases according to the Euler equation $S = S_0 e^{ct}$

The product winding tension depends on the coefficient $f$ of the tape friction along the path guides, the total angle of coverage by the tape of all path guides $\alpha$ and tension $S_0$ when the tape enters the winding path.

Thus, the tensioners with wet winding can be constructed having regard to the effects of the friction coefficient changing and the change of the angle of coverage by the tape of the path guides.

It is theoretically possible to change the friction coefficient $f$ during the winding [17], but this is irrational.

The tension control implementation is easier by changing the angle $\alpha$ of coverage by the tape of the path guides. For this purpose, the rollers can be installed in the winding path one after the other. When changing the position of the rollers in space, the angle of coverage changes, and it follows that the tape tension when the winding on the product changes.

There are many designs of these tensioners. Figures 2–4 show the designs of the most commonly used tensioners [1, 13, 15].

The wet tape 1 (Figure 2 a) passes through non-rotating rollers 2 and 3, with the rollers 2 fixed and the roller 3 moving up and down using a special device. This changes the coverage angle of the rollers 2 and 3 by the tape, and therefore, the tension at the outlet of the device changes. Figures 2 b–c show the non-rotating rollers 2 that change their position by rotating along axis 3. The changing of the rollers position changes the angle of their coverage by the tape 1 and therefore, the tension at the outlet of the
device. Figure 2 d shows the tensioner with the fixed bottom roller 2 and the top roller that moves along the tensioner on the guide 4.

Figure 2. Tensioners designs for the wet winding

Figure 3. Scheme of a tensioner consisting of two interconnected "eights" with the "angular movement of rollers" (1 – is the elastic tape; 2 – are the fixed rollers; 3 – is the center of “eights” rotation; 4 – are three connected gears; 5 – is the pinion rotation axis)
Figure 4 Scheme of a tensioner with the “linear movement” of the roller and an additional pneumatic drive

Figure 4 shows the scheme of a tensioner with the “linear movement” of the roller and an additional pneumatic drive, where 1 – is the elastic tape; 2, 3 – are the tension rollers; 4 – is the reducer; 5 – is the electric drive; 6 – is the pneumatic cylinder rod; 7; 8 – is the gear connected with the reducer 4; 9 – is the rack where the gear 8 rolls on.

When winding "Cylinder" products they move the tensioner fixed rollers only to compensate for disturbing influences, which are caused by uneven tape adhesion when it passes the fixed guides of the winding machine. These disturbing influences are not considerable. The wet material tension level causes the main moment developed by the electric motor.

To determine the relationship between the required dynamic characteristics of the tensioner and its parameters (gear ratio and other elements of the kinematic mechanism \( k_u \)) we must put the movement time of the tensioner from one extreme position to another one \( T_{cyc} \). The dependence of the displacement angle on the engine rotating speed \( \Omega_m \) is as follows,

\[
\Delta \alpha_\text{max} = \int_0^{T_{cyc}} \frac{\Omega_m(t)}{k_u} dt.
\]

For the fastest movement, the \( \Omega_m(t) \) must be constant and equal to its maximum value, therefore we have:

\[
\Delta \alpha_\text{max} = \frac{\Omega_{m,max}}{k_u} T_{cyc}.
\]

The gear ratio of the reducer must be less than or equal to:
The tensioner with the “angular movement” of rollers has the self-braking gearbox with the gear ratio $k_m$. The tensioner with the “linear movement” has the self-braking gearbox and the gear connected with a gearbox and rolling on a rack. In this case, the overall gear ratio is $k_m \cdot r_G$.

The driving device applied force $M_c$ (fig. 5) depends on the tensioner and is calculated by the expressions (4) or (5). The rated torque of the selected motor must be greater than the maximum $M_c$ value.

For the tensioner with the “angular movement” of rollers the maximum $M_c$ value equal to:

$$M_{c,\text{max}} = \frac{\arccos\left(\frac{2r}{L}\right) \cdot \left[ S_1 + S_0 \cdot e^{\frac{2}{L}} \cdot \frac{r}{L} \right]}{k_m} \cdot (L + r) + M_T.$$  

(4)

For the tensioner with the “linear movement” it equal to:

$$M_{c,\text{max}} = \frac{f \cdot \arccos\left(\frac{2r}{L}\right) \cdot \left[ S_1 + S_0 \cdot e^{\frac{2}{L}} \cdot \frac{r}{L} \right]}{k_m} \cdot r_g + M_T.$$  

(5)

with two “eights”:

$$M_{c,\text{max}} = \frac{\left[ S_1 - S_0 + S_0 \cdot e^{\frac{3}{L}} \cdot \frac{3r}{L} - S_0 \cdot e^{\frac{2}{L}} \cdot \frac{r}{L} \right]}{k_m} \cdot (L + r) + M_T.$$  

(6)

The fulfillment of the following relation is the condition for choosing the engine power proceeding from the maximum shaft torque:

$$M_D = k_s \cdot M_{c,\text{max}},$$  

(7)

where, $k_s$ – is the safety factor.

When winding the complex geometric shapes products, the total coverage angle of the guides by the wet tape changes significantly due to the work of the stacker, hereupon the winding tape tension changes. The tensioner neutralizes this disturbing effect by leaving permanent the total coverage angle of the guides by the tape. The drive must provide such a conveying speed of the actuator, so that the tension does not change when the tape lading on the product. This speed is determined by the tensioner design, the size of the gearbox installed between the actuator and the electric motor and the maximum rotation velocity of the motor shaft. Furthermore, the drive must provide the required shaft torque of the actuator and the required.

To take into account the influence of the stacker on the machine operation in the sources [15, 17-20] they offered an "extended kinematic scheme", which includes the entire path of the winding equipment. Based on this kinematic scheme we can receive the winding tension force changing graph for the calculated trajectories of the spreader, and consequently, the tensioner actuator rollers rotation angle graph from time $\varphi(t) = \Lambda(t)$ for the tensioners of the types b, c figure 2 or the rollers moving along the
guides $x(t) = \Lambda(t)$ the types a, d figures 2 and 3. We use these graphs and corresponding kinematic schemes of the tensioners to obtain the dynamic torque of the drive.

The calculation sequence is given below.

Using the known function $\Lambda(t)$, we calculate $\alpha(t) = \Lambda(t)$ for the selected actuator.

Using the known $\alpha(t)$, we calculate $M_c(t) = \Lambda2(t)$.

Using the known $\alpha(t)$, we calculate $\Omega_d(t)$.

Using the basic equation of the drive dynamics, we calculate the drive dynamic torque developed in the process of the actuator rollers moving:

$$M_D = M_c(t) + J_d \frac{d\Omega_d}{dt}.$$  
(8)

We calculate the equivalent motor torque:

$$M_{eq} = \sqrt{\frac{1}{t_{cyc}} \int_{0}^{t_{cyc}} M^2_D(t) dt},$$  
(9)

where $t_{cyc}$ – is the time of one complete cycle of the machine (of one tape turn on the product).

4. Conclusions

1) We investigated the static and dynamic components influence on the drive equivalent torque. We concluded that the dynamic component in these tensioners is small. Therefore, in preliminary calculations, we can neglect it and use a safety factor equal to 1.1-1.2.

2) We investigated the various types of tensioners. In the tensioners, it is more advisable to use the "angular movement" of the rollers and two sequentially included "eights", connected in a single mechanism.

3) The use of a low efficiency self-braking worm gearbox does not significantly increase the drive power of the tensioner. However, in case of idle time the engine is not in the power consumption mode, no requirement for a standby-load mode drive. This allows the use of cheaper actuator motors.

4) The new method of the tensioners drive power choosing takes into account the maximum static moment of the actuator when winding the product.

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