Selecting the engine power while the dry winding of the composite material

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Abstract. Products made by winding of composite materials are very prevalent in different industries. The product range is expanding and the quality requirements are constantly growing. The quality characteristics of future products are greatly dependent on the properties of the tension control system. The settings and type of the controller and tensioners drive power determine the system operation. Typically, the choice of motor takes into account only the specified maximum tension of the composite tape and does not consider some specific features of the wound product. However, while winding of complex geometric shape products, dynamic torques cause an additional heating of the engine. This article describe the technique of tensioners drive power selection. This technique considers winding features of composite products having complex shape. The products are qualified by form depending on the dynamic torque when winding the product.

1. Introduction
The products made of composite material by winding method have unique chemical and mechanical characteristics and they are applied in many industries: in spacecraft and rockets, in aviation, in the chemical industry [1-7]. There are two widely used types of winding: dry winding and wet winding.

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

Figure 1. Dry winding path
where 1 is a bobbin with a “dry” tape, 2 is a brake device, that slows down the bobbin with a tape; 3 is the “dry” tape; 4 are the rubber brake rollers of a tensioner; 5 is a tensioner drive; 6 is a tension meter; 7 is a rotating roller of the winding path; 8 is a wound product; 9 is a drive.

The change of a torque on the drive shaft 5 provides the predetermined tension $S_1$ in the winding area. The tape leaves the bobbin 1 and has a tension $S_0$. The torque on the brake device shaft 2 and the bobbin radius 1 determine this tension.

In connection with the improvement of winding technology, the requirements began more stringent for a drive performance, a changing operating modes accounting, and an adaptability when changing winding modes [8-13].

2. Formulation of the problem

Until now, they have determined the tensioner drive power by the method of equivalent current, torque, power or by the empirical expressions without taking into account the features of the wound product. In such a case, the motor must provide a predetermined maximum tension value at a maximum winding speed. However, the shape of the wound product influence on the dynamic torques in the electric drive. The checking of the dynamic torques in the selected drive could be done only when modeling a tension control system. If the drive was chosen incorrectly, it was necessary to repeat the whole calculation again.

A quite extensive literature is devoted to the electric drive power of different tensioners. However, in all the researches there is no accounting for the winding composite products specifics. For example, there is no accounting of the dynamic torque causing additional engine heat when winding a product of complex geometric shape.

3. Theoretical part

To take into account specific features when choosing a tensioner electric motor we propose the methodology for selecting the tensioners drive power for the dry winding. We offer the following sequence when choosing the power:

- to calculate drive power, taking into account the worst static winding conditions, the maximum tape winding tension, the minimum preliminary pretension and the maximum linear winding speed;
- to find equivalent torque of the engine on selected winding equipment for winding the most complex product and to check whether the selected engine provides heating conditions. If not then we must choose the engine of higher power.

Since the heating is the condition for choosing the engine, the RMS torque per working cycle is advisable to choose as the equivalent torque.

Let us consider sequentially the proposed methodology stages.

To determine the drive power 5 (fig. 1) in static mode we used expressions [1,14-20] that connect the moment on the shaft of the tensioner in the steady state and its rotating speed:

$$M_{c1} = \frac{r_1 \cdot \eta_1}{i_1} (S_1 - S_0);$$

$$\Omega_1 = \frac{i_1}{r_1} \Omega_1,$$

(1)

where $M_{c1}$ – is the static torque on the drive shaft of the tensioner; $v_1$ – is the linear velocity of the material entry into the deformation zone; $r_1$ – is the brake roller radius; $\eta_1$ – is the gearbox efficiency; $i_1$ – is the gear ratio of the reducer located between the tensioner rollers and the drive (there is no a reducer on the figure 1); $S_1$ – is the tape tension value when winding on a product; $S_0$ – is the tape tension value when winding on bobbins; $\Omega_1$ – is the angular rotation speed of the tensioner drive.

Obviously, the tensioner drive power in static mode is:
The obtained expressions (1) and (2) allow calculating the tensioner drive power in static mode and maximum torque on the drive shaft but do not take into account its dynamics.

At the second stage, we check the selected engine using the equivalent torque method. Obviously, the tensioner engine torque will be the sum of the static torque and dynamic torque. We can use expression (2) to calculate the static torque. The dynamic torque is related to the change in the linear speed of the winding material.

The analysis of products made by winding showed that they are divided into three groups.

In the manufacture of first-group products, the diameter of the winding changes slightly and we can assume that the linear winding speed remains constant. Let us call these products “Cylinder” (Fig. 2 a).

During winding, the drive very rarely develops the dynamic moments. Equipment stops to change tape spools or under some unforeseen circumstances. Equipment stops to change tape spools or under some unforeseen circumstances. Such situations arise, as a rule, once every few hours. Therefore, we do not consider them. Thus, the condition for choosing a motor for heating is the following:

\[
P_1 = M_{c1} \cdot \Omega_i = (S_{1\text{max}} - S_{0\text{min}}) \cdot \nu_{1\text{max}} \cdot \eta_i,
\]

where, \(S_{1\text{max}}\) is the maximum tape tension value when winding on a product (maximum tape tension value of a specific winding machine); \(S_{0\text{min}}\) is the minimum tension value when the tape winding from a bobbin; \(\nu_{1\text{max}}\) is the maximum tape speed.

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\[
M_D \geq M_{Eq} = M_{c1} = \frac{(S_{1\text{max}} - S_{0\text{min}}) \cdot r_1 \cdot \eta_i}{t_1}
\]

Figure 2. The shape of some winding products

In the manufacture of second-group products, the diameter of the winding changes significantly (Fig. 2 b) - up to three times or more in one run. Let us call these products “Cone”. Figure 3 shows the tachogram of the drive operation with a constant speed when winding such a product.
Figure 3. The linear tape speed when winding "Cone" shape products

The carriage of the machine moves along the product at a speed $v_{car}$. The winding time of a section

$$t_i = \frac{L_i}{v_{car}}$$

of length $L_i$ is equal to $v_{car}$. In the section $L_1$ and $L_2$, the tape speed increases when the carriage moves to the right and decreases when the carriage moves to the left.

The speed of the tape in different sections of the product is as follows:

$$v_{lin,\min} = \Omega_m \cdot h_{2\min}$$

$$v_{lin} = \Omega_m \left[h_{2\min} + \frac{h_{2\max} - h_{2\min}}{L_2} \cdot v_{car} \cdot (t - t_1)\right]$$

$$v_{lin,\max} = \Omega_m \cdot h_{2\max},$$

where, $\Omega_m$ is the mandrel rotation speed.

We calculate the drive moment in all winding sections taking into account the basic equation of drive dynamics $M - M_c = J_1 \frac{d\Omega}{dt}$. Then we get in each section:

- in the first section from 0 to $t_1$:

$$M_{c1} = \frac{(S_{1\max} - S_{0\min}) \cdot r_i \cdot \eta_i}{i_1}$$

- in the second section from $t_1$ to $t_2$ we replace the derivative with an increment:

$$M_{c2} = J_1 \frac{(v_{lin,\max} - v_{lin,\min}) \cdot i_1}{(t_2 - t_1) \cdot r_i} + \frac{(S_{1\max} - S_{0\min}) \cdot r_i \cdot \eta_i}{i_1}$$

- in the third section from $t_2$ to $t_3$:

$$M_{c3} = \frac{(S_{1\max} - S_{0\min}) \cdot r_i \cdot \eta_i}{i_1}$$

We get:

$$M_D \geq M_{EQ} = \sqrt{\frac{M_{c1}^2 t_1 + M_{c2}^2 (t_2 - t_1) + M_{c3}^2 (t_3 - t_2)}{t_4}}$$
The condition for the correct selection of drive power is:

\[ M_D \geq M_{EQ} \]

Obviously, the equivalent motor torque depends on the size and shape of the wound product, the machine dimensions and the mandrel rotation speed. Therefore, in the synthesis of the tension control system, it is necessary to analyze the entire range of wound products, to choose products with maximum dynamic engine torques when winding and to calculate the equivalent engine torque for such a product.

4. Conclusions
1. The selection of the tensioners drive power based only on the static winding torque has disadvantages and leads to the engine overheating.
2. The new method of the tensioners drive power selecting takes into account the dynamic torque associated with a change in the linear speed of the composite products winding.
3. The proposed classification of wound products based on their shape depends on the dynamic torques that occur in the electric drive when their winding.
4. To prevent overheating of the tensioner motor when winding complex shape products, the drive power determined using the proposed methodology must exceed 2.5 times the drive power determined only with a static load.

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