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SIMULATION MODEL OF THE WEAVING MACHINE "CAMEL" AND SELECTION OF THE SUFFICIENT DRIVING MOTOR

SIMULAČNÍ MODEL TKACÍHO STAVU "CAMEL" A SPRÁVNÝ VÝBĚR HNACÍHO MOTORU

Abstract

The paper deals with the mathematical model of the waving machine CAMEL. This machine consists of many moving parts (rotational and translational movements), belts, flexible elements and therefore it is very complex. CAMEL uses servomotors working in electronic cam regime. It means that the actual angular velocity of the rotor is not constant and therefore it is really important to reduce the moment of inertia of rotating elements. The inertia of the rotor of the drive is very important too. Existing simulation model can help to choose the optimal drive of the machine. It also allows selecting the best displacement laws for different speeds (rpm) in order to decrease the effective torque which is proportional to the heating of the servomotor.

Abstrakt

Článek se zabývá matematickým modelem tkacího stroje CAMEL. Ten obsahuje mnoho pohybujících se součástí, vykonávající buď rotační anebo translační pohyb, řemeny, poddajné členy a proto se pro matematický popis stává poměrně složitý. CAMEL používá servomotory v režimu elektronické vačky, což zjednodušeně řečeno znamená, že motory úhlová rychlost motoru není konstantní, ale v rámci periody zrychluje nebo zpomaluje na základě předpisu neboli zdvihové závislosti. Hodnota momentu setrvačnosti rotoru je tedy také velmi důležitá. Matematický model mechanické části stroje pomůže vybrat optimální motor, který dokáže obsloužit požadovaný pracovní cyklus, může také pomoci s výběrem optimální zdvihové závislosti a to vše za účelem snížení efektivního momentu motoru, který je přímo úměrný zahřátí nebo přehřátí motoru.

Keywords

electronic cam, weaving machine, optimal displacement law, effective torque

1 INTRODUCTION

The weaving machine CAMEL is a complex machine with several servomotors. These servomotors mostly represent the electronic cams. The most critical is the "main" servodrive which is supposed to ensure the major movement of the machine. The major movement consists of slay motion and shedding motion which are coupled. The reciprocating motions are ensured by the crank mechanisms. The mechanism uses the accumulation of energy using the flexible link between the

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slay and the frame. The flexible link works like a spring and helps the reversing of the slay at the dead centre position. The spring accumulates the kinetic energy and helps the decelerating of the slay and then release the energy during the accelerating. The kinematic schema of mechanism is in Fig. 1.

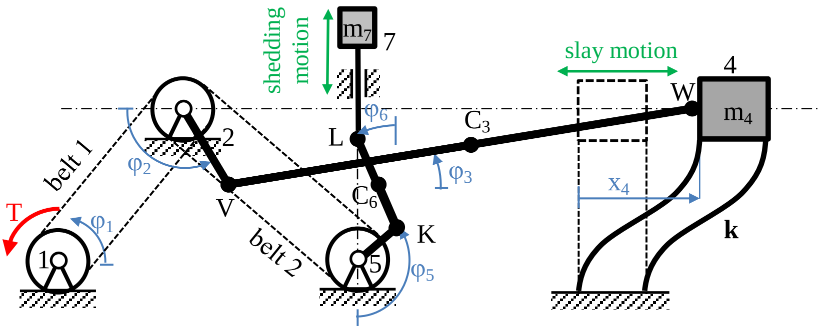

Fig. 1 Schema of the major mechanism of CAMEL weaving machine

2 DYNAMIC MODEL

The dynamic equations of each body can be obtained. Mechanism described in Fig. 1 can be described by 11 dynamic equations [2]. If the belts are considered as rigid the bodies 1, 2 and 5 became one body and the number of equation reduces to 9. Another reduction can be done, when both crank mechanisms are considered as one and mass m_7 becomes a part of the mass m_4. The number of dynamic equations is reduced to 5 but it can be dangerous that we lose some dynamic properties. When the displacement law $\varphi_1(t)$ is strictly fixed the positions, velocities and accelerations of all bodies can be computed in certain time moments. In these moments the driving torque can be computed solving the set of dynamic equations. If the flexibilities of the belts are not considered the inverse dynamics is easy.

When only one crank mechanism is considered the dynamic equations in matrix form (6) can be solved by inverting matrix A and using formula (7). The reaction forces in joints V and W and the requested torque T are calculated.

$$\varphi_2 = \varphi_1 \tag{1}$$

$$\varphi_3 = \arcsin\left(\frac{l_2}{l_3}\sin\varphi_2\right) \tag{2}$$

$$x_3 = -l_2\cos\varphi_2 + \frac{l_3}{2}\cos\varphi_3 \tag{3}$$

$$y_3 = -l_2\sin\varphi_2 + \frac{l_3}{2}\sin\varphi_3 \tag{4}$$

$$x_4 = -l_2\cos\varphi_2 + l_3\cos\varphi_3 + l_2 - l_3 \tag{5}$$

$$\underbrace{\begin{bmatrix} l_2\sin\varphi_2 & -l_2\cos\varphi_2 & 0 & 0 & 1 \\ -1 & 0 & -1 & 0 & 0 \\ 0 & -1 & 0 & -1 & 0 \\ -\frac{l_3}{2}\sin\varphi_3 & \frac{l_3}{2}\cos\varphi_3 & \frac{l_3}{2}\sin\varphi_3 & -\frac{l_3}{2}\cos\varphi_3 & 0 \\ 0 & 0 & 1 & 0 & 0 \end{bmatrix}}_{A} \underbrace{\begin{bmatrix} R_{Vx} \\ R_{Vy} \\ R_{Wx} \\ R_{Wy} \\ T \end{bmatrix}}_{X} = \underbrace{\begin{bmatrix} (I_1 + I_2 + I_5)\ddot{\varphi}_2 \\ (m_3 + m_6)\ddot{x}_{C_3} \\ (m_3 + m_6)\ddot{y}_{C_3} \\ (I_{3C_3} + I_{6C_6})\ddot{\varphi}_3 \\ (m_4 + m_7)\ddot{x}_4 + kx_4 \end{bmatrix}}_{B} \tag{6}$$

l_i – length of i-th link [m]

φ_i – angular position of i-th link [rad]

x_P, y_P – x,y position of the point P [m]

I_{iP} – inertia of i-th body to the point P [kg·m²]

m_i – mass of body i [kg]

R_i – reaction force in the i-th joint [N]

k – stiffness of the slay [N/m]

T – driving torque [N]

$$X = A^{-1}B \tag{7}$$

When both crank mechanisms are considered, the kinematics of bodies 5, 6 and 7 is needed.

$$\varphi_5 = \varphi_2 \tag{8}$$

$$\varphi_6 = \arcsin\left(\frac{l_5}{l_6}\sin\varphi_5\right) \tag{9}$$

$$y_7 = -l_5\cos\varphi_5 + l_6\cos\varphi_6 + l_5 - l_6 \tag{10}$$

The driving torque and the reaction forces are calculated using the matrix formula (11). It is more complex but it can be reduced when reaction forces are not needed to compute.

$$\begin{bmatrix} l_2 s\varphi_2 & -l_2 c\varphi_2 & 0 & 0 & -l_5 c\varphi_5 & l_5 s\varphi_5 & 0 & 0 & 1 \\ -1 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & -1 & 0 & 0 & 0 & 0 & 0 \\ -\frac{l_3}{2}s\varphi_3 & \frac{l_3}{2}c\varphi_3 & \frac{l_3}{2}s\varphi_3 & -\frac{l_3}{2}c\varphi_3 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{l_6}{2}c\varphi_6 & -\frac{l_6}{2}s\varphi_6 & -\frac{l_6}{2}c\varphi_6 & \frac{l_6}{2}s\varphi_6 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} R_{Vx} \\ R_{Vy} \\ R_{Wx} \\ R_{Wy} \\ R_{Kx} \\ R_{Ky} \\ R_{Lx} \\ R_{Ly} \\ T \end{bmatrix} = \begin{bmatrix} I_{2O}\ddot{\varphi}_2 \\ m_3\ddot{x}_{C_3} \\ m_3\ddot{y}_{C_3} \\ I_{3C_3}\ddot{\varphi}_3 \\ m_4\ddot{x}_4 + kx_4 \\ m_6\left(\ddot{y}_{C_6} + g\right) \\ m_6\ddot{x}_{C_6} \\ I_{6C_6}\ddot{\varphi}_6 \\ m_7\left(\ddot{y}_7 + g\right) \end{bmatrix} \tag{11}$$

3 SIMULATION AND EXPERIMENTAL RESULTS COMPARISON

One existing weaving machine CAMEL was selected, all threads and other weaving material was removed in order to test the mechanical part of the machine only. The servomotor Yaskawa SGMSV-30D (3kW capacity, rated torque 9.8 Nm, peak torque 29.4 Nm, rated speed 3000 rpm, max. speed 5000 rpm) SGMG 3kW [3] and planetary gearbox Stöber [4] (gear ratio i=4) were used. The gear ratio is not used in the equations. The mathematical model does not take the gearbox into account. The computed required torque is then divided by the gear ratio. The required positions, velocities and accelerations are multiplied by the gear ratio.

The displacement law is computed in order to satisfy the weaving process. The slay position has to be higher than 28mm at least for picking angle of 210 deg of independent master τ (12). This is due to the weaving technology. The displacement law is the dependence of the position φ_1 on master position τ (13) and it is designed in order to satisfy the picking angle. In this case the inclined sinus line [1] modified by VDI 2143 [5] is used. The displacement law is displayed in Fig. 2 where the displacement $x_4(\tau)$ is displayed too.

$$\tau = 60 \cdot n \cdot t \tag{12}$$

τ – independent master position [deg]

n – angular velocity of the machine [rpm]

t – time [s]

$$\varphi_1 = \varphi_1(\tau) \tag{12}$$

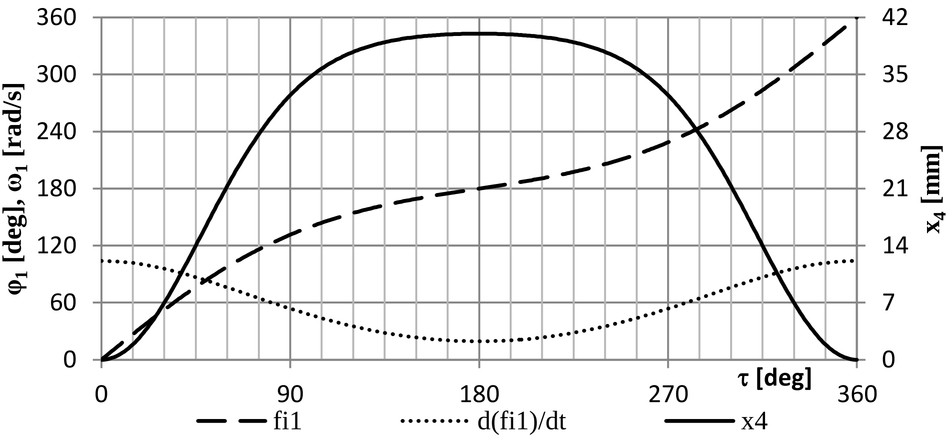

Fig. 2 Displacement law

The mechanical properties of the CAMEL machine were measured and used in the simulation model. Because the gearbox was used all masses on the input side (inertia of the rotor, inertia if the gearbox) were transformed to the output side using formula (13).

$$I_1 = I_{2O} + I_{5O} + i^2\left(I_{rot} + I_{gear}\right) \tag{13}$$

I_{rot} – inertia of the rotor [kgm^2]

I_{gear} – inertia of the gearbox [kgm^2]

i – gear ratio of the gearbox [-]

The values used in simulation:

I_{rot} = 0.0007 kgm^2, I_{gear} = 0.0003 kgm^2, I_2 = 0.018537 kgm^2

m_3 = 1.25kg, m_4 = 17kg, m_6 = 2.5kg, m_7 = 3kg, k = 130000 N/m

l_2 = 20mm, l_3 = 145mm, l_5 = 30mm, l_6 = 290mm

The simulation using equation (6) and equation (11) are compared with the measured data too. The results are compared for 3 different speeds (350rpm, 450rpm, 550rpm) in figures Fig.3-Fig.5. The difference is obvious. The influence of the shedding is evident because its stroke is 50% higher than the stroke of the slay. The difference in the computed effective torque is not too high (Tab. 1).

Tab. 1 Effective torque comparison

Speed [rpm]	Effective torque [Nm] 1-crank model	Effective torque [Nm] 2-crank model	Measured effective torque [Nm]
350	5.26	5.16	4.96
450	3.29	3.50	3.68
550	7.2	7.98	7.45

The compared torques are recomputed to the motor side of the gearbox and displayed above the position of the input shaft (φ_1). The required torque for accelerating and decelerating of the rotor itself is displayed too. The inertia of the rotor has really big influence. The exact simulation model helps the verifying of selected the servomotor. It can even help during the machine design.

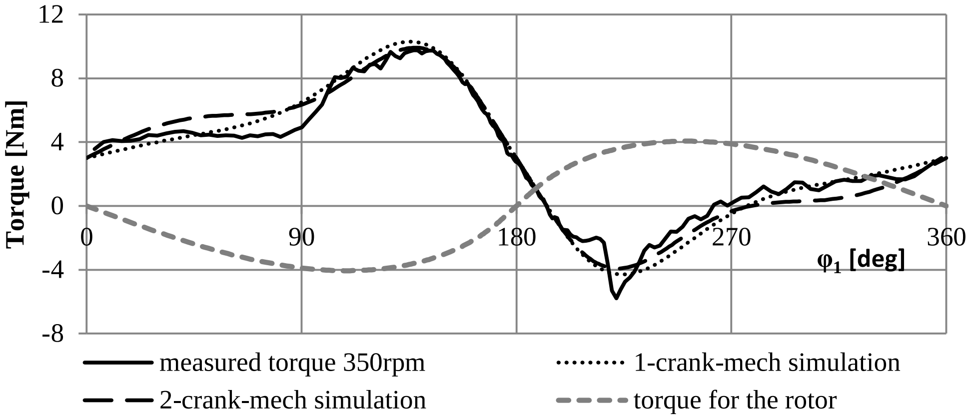

Fig. 3 Simulation/reality comparison – speed 350rpm

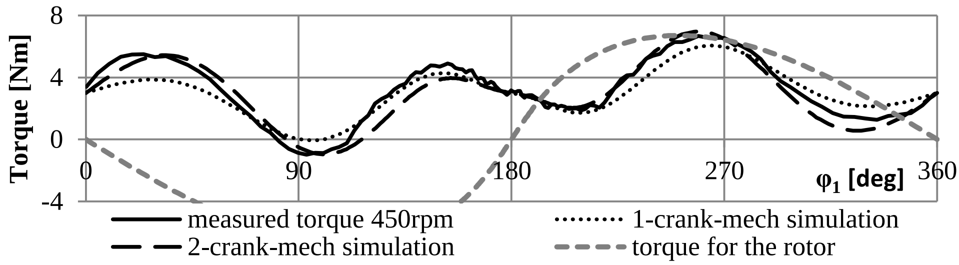

Fig. 4 Simulation/reality comparison – speed 450rpm

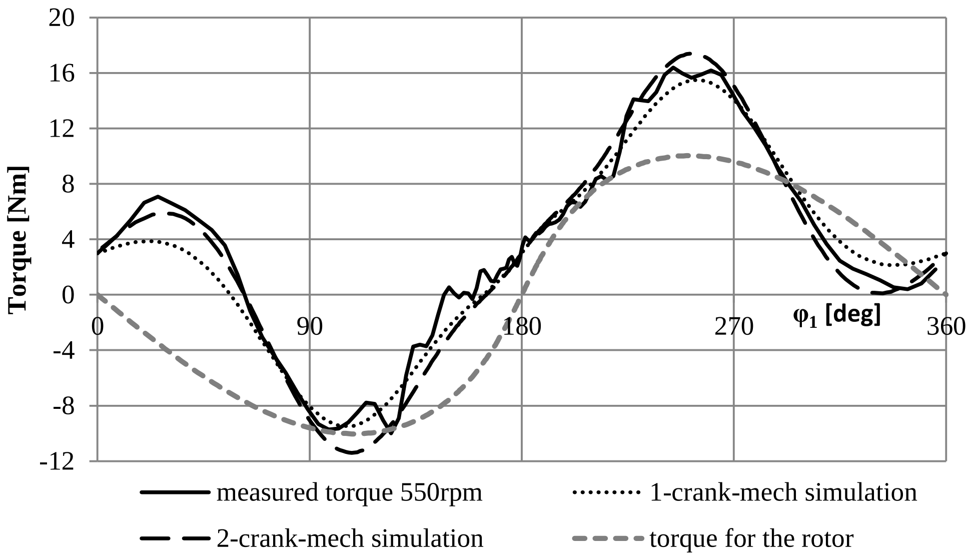

Fig. 5 Simulation/reality comparison – speed 550rpm

4 OPTIMAL DISPLACEMENT LAW

When the main drive is selected and all dimensions, masses, springs are known the displacement law can be optimized too. It is optimized in order to minimize the effective torque and uses already described simulation model. The parametric description of the displacement law and its right selection is very important. The parameters can be optimized using global optimization of parameters which uses the simulation model of the machine too.

Let consider the displacement law (14) and (15) described with parameters κ and η can be the right one. The cost function for the optimization is directly the computed effective torque (16). The optimization is looking for such parameters the cost function f_{cost} has minimal value with.

$$\varphi = \mu + \frac{\eta}{2\pi}\sin(2\pi\mu) \tag{14}$$

$$\mu = \tau - \frac{\kappa}{2\pi}\frac{\sin(2\pi\tau)}{1+\kappa\cos(2\pi\tau)} \tag{15}$$

$$f_{\cos t} = T_{eff} = \sqrt{\frac{n}{60}\int_{t_{start}}^{t_{start}+\frac{60}{n}} \left(T(t,\kappa,\eta)\right)^2 dt} \tag{16}$$

It is possible to optimize the displacement law for every machine speed. It means to find the parameters κ and η which satisfies the desired picking angle and minimizes the computed driving torque for each machine speed.

5 CONCLUSIONS

The exact mathematical model of mechanical system is very important in order to predict the dynamic behaving of the mechanism. The suitable servomotor can be selected even the suitable displacement law can be computed. The pre-computed drive torque can be used as a feed forward torque in the servodrive regulation which helps to reduce the position error and the satisfying the weaving technology.

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