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DYNAMIC MODEL OF WEAVING MACHINE CAMEL INCLUDING DRIVES

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Abstract

The paper deals with the mathematical model of the weaving machine CAMEL dynamics. That machine consists of many moving parts (rotational and translational), belts, flexible elements and therefore it is fairly complex. CAMEL uses several servomotors working in electronic cam regime. It means that the angular velocity of the rotor is varying in time regarding the prescribed profile. Therefore the moment of inertia of rotating elements is really important parameter. It affects the dynamics and then torque requirements of the motor. The inertia of the rotor is very important too. Light-rotor motors have better dynamics. Existing mathematical model of the machine can help to choose the optimal drive of the machine. It also allows the selecting of the optimal displacement laws for different speeds (rpm) in order to decrease the effective torque which is proportional to the heating of servomotor.

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 crank mechanisms ensure the reciprocating motions. The mechanism utilizes the accumulation of energy using the flexible link between the slay and the frame. The flexible link works like a spring and helps the reversing of the slay to 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.

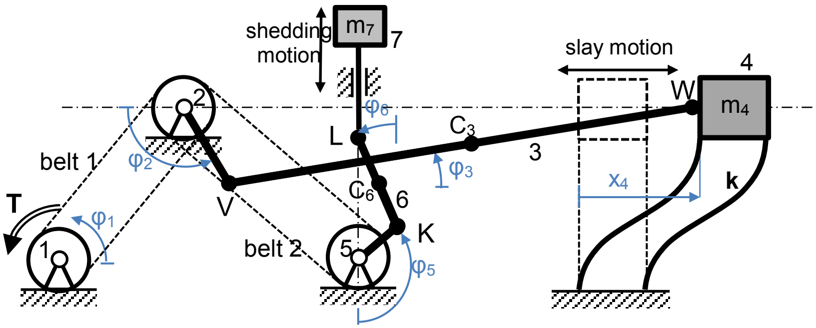

Fig. 1: The major mechanism of CAMEL weaving machine schema

2 Dynamic model

The dynamic equations of each body can be obtained [2]. Mechanism described in Fig. 1 can be described by 11 dynamic equations. If the belts are considered as rigid no mass elements, the bodies 1, 2 and 5 become one body and the number of equations reduces to 9. Another reduction can be arranged considering one crank mechanism only. When both crank mechanisms are considered as one (1-crank approach) 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 because some important dynamic properties can be lost. When the displacement law $\varphi_1(t)$ is strictly fixed, the positions, velocities and accelerations of all bodies can be computed in certain time. Therefore the velocity profile and acceleration profile are considered as known too. 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 (2.6) can be solved by inverting matrix A and using formula (2.7). The reaction forces in joints V and W and the requested torque T are calculated.

$$\varphi_2 = \varphi_1 \tag{2.1}$$

$$\varphi_3 = \arcsin\left(\frac{l_2}{l_3}\sin\varphi_2\right) \tag{2.2}$$

$$x_3 = -l_2\cos\varphi_2 + \frac{l_3}{2}\cos\varphi_3 \tag{2.3}$$

$$y_3 = -l_2\sin\varphi_2 + \frac{l_3}{2}\sin\varphi_3 \tag{2.4}$$

$$x_4 = -l_2\cos\varphi_2 + l_3\cos\varphi_3 + l_2 - l_3 \tag{2.5}$$

$$\underbrace{\begin{bmatrix} l_2\sin\varphi_2 & -l_2\cos\varphi_2 & 0 & 0 & 1 \\ -1 & 0 & -1 & 0 & 0 \\ 0 & -1 & 0 & -1 & 0 \\ -\frac{l_3}{2}\sin\varphi_3 & \frac{l_3}{2}\cos\varphi_3 & \frac{l_3}{2}\sin\varphi_3 & -\frac{l_3}{2}\cos\varphi_3 & 0 \\ 0 & 0 & 1 & 0 & 0 \end{bmatrix}}_{A} \underbrace{\begin{bmatrix} R_{Vx} \\ R_{Vy} \\ R_{Wx} \\ R_{Wy} \\ T \end{bmatrix}}_{X} = \underbrace{\begin{bmatrix} (I_1 + I_2 + I_5)\ddot{\varphi}_2 \\ (m_3 + m_6)\ddot{x}_{C_3} \\ (m_3 + m_6)\ddot{y}_{C_3} \\ (I_{3C_3} + I_{6C_6})\ddot{\varphi}_3 \\ (m_4 + m_7)\ddot{x}_4 + kx_4 \end{bmatrix}}_{B}, \tag{2.6}$$

where:

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·m^2]
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]

$$X = A^{-1}B \tag{2.7}$$

When both crank mechanisms are considered, the kinematics of bodies 5, 6 and 7 is needed. Each crank mechanism has its own dynamics and the influence each other because they are coupled through the belt 2. When the lengths of the cranks are not same it is better to use 2-cranks approach. Masses m_7 and m_4 have different strokes in the equal period, therefore the accelerations are not equal and dynamic forces are not equal.

$$\varphi_5 = \varphi_2 \tag{2.8}$$

$$\varphi_6 = \arcsin\left(\frac{l_5}{l_6}\sin\varphi_5\right) \tag{2.9}$$

$$y_7 = -l_5\cos\varphi_5 + l_6\cos\varphi_6 + l_5 - l_6 \tag{2.10}$$

The driving torque and the reaction forces are calculated using the matrix formula (2.11). It is more complex then (2.6) but it can be reduced when reaction forces are not needed to compute.

$$\begin{bmatrix} l_2 s\varphi_2 & -l_2 c\varphi_2 & 0 & 0 & -l_5 c\varphi_5 & l_5 s\varphi_5 & 0 & 0 & 1 \\ -1 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & -1 & 0 & 0 & 0 & 0 & 0 \\ -\frac{l_3}{2}s\varphi_3 & \frac{l_3}{2}c\varphi_3 & \frac{l_3}{2}s\varphi_3 & -\frac{l_3}{2}c\varphi_3 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & -1 & 0 & -1 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{l_6}{2}c\varphi_6 & -\frac{l_6}{2}s\varphi_6 & -\frac{l_6}{2}c\varphi_6 & \frac{l_6}{2}s\varphi_6 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} R_{Vx} \\ R_{Vy} \\ R_{Wx} \\ R_{Wy} \\ R_{Kx} \\ R_{Ky} \\ R_{Lx} \\ R_{Ly} \\ T \end{bmatrix} = \begin{bmatrix} I_{2O}\ddot{\varphi}_2 \\ m_3\ddot{x}_{C_3} \\ m_3\ddot{y}_{C_3} \\ I_{3C_3}\ddot{\varphi}_3 \\ m_4\ddot{x}_4 + kx_4 \\ m_6\left(\ddot{y}_{C_6} + g\right) \\ m_6\ddot{x}_{C_6} \\ I_{6C_6}\ddot{\varphi}_6 \\ m_7\left(\ddot{y}_7 + g\right) \end{bmatrix} \tag{2.11}$$

3 Simulation and experiments

One existing weaving machine CAMEL was selected. All threads and other weaving material were 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 in order to drive the main mechanism. The gearbox is not considered in the equations, the mathematical model does not take it 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 28 mm at least for picking angle of 210 deg of independent master τ (3.1). This is due to the weaving technology and therefore the mechanism cannot be driven by constant velocity motor. The displacement law is actually the dependence of the position φ_1 on master position τ (3.2) and it is designed in order to satisfy the picking angle and minimal effective torque consumption. 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.

$$\tau = 60 \cdot n \cdot t \quad (3.1)$$

where:

τ – independent master position [deg]
n – angular velocity of the machine [rpm]
t – time [s]

$$\varphi_1 = \varphi_1(\tau) \quad (3.2)$$

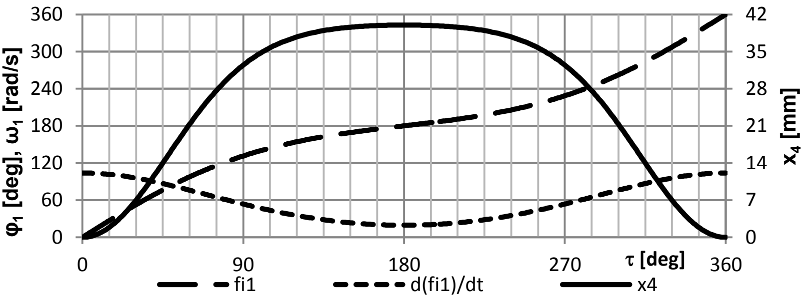

Fig. 2: Displacement law

The mechanical properties of the CAMEL machine were measured experimentally and then 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 (3.3).

$$I_1 = I_{2O} + I_{5O} + i^2\left(I_{rot} + I_{gear}\right) \quad (3.3)$$

where:

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 (2.6) and equation (2.11) are compared with the measured data too. The results are compared for 3 different speeds (350 rpm, 450 rpm, 550 rpm) 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 1-crank model [Nm]	Effective torque 2-cranks model [Nm]	Measured effective torque [Nm]
350	5.26	5.16	4.96
450	3.29	3.50	3.68
550	7.20	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 inertia of the rotor has really high influence. The exact simulation model helps the verifying of selected the servomotor. It can even help during the machine design.

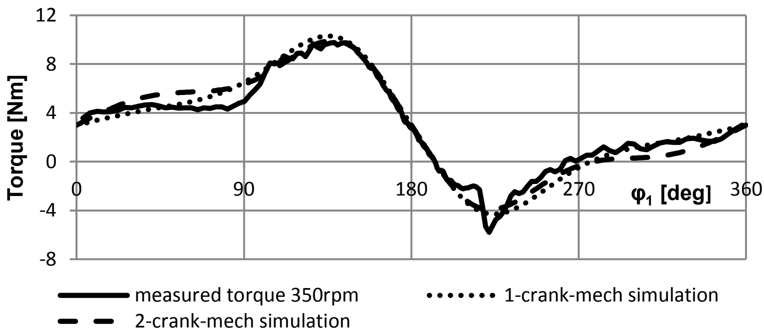

Fig. 3: Simulation/reality comparison – speed 350rpm

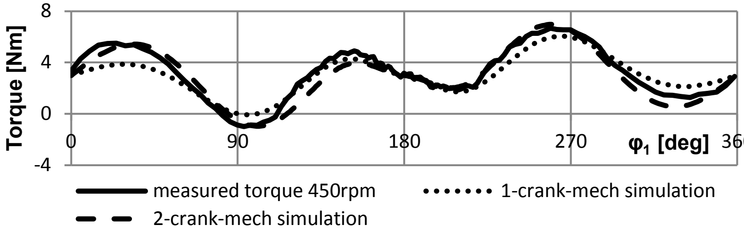

Fig. 4: Simulation/reality comparison – speed 450rpm

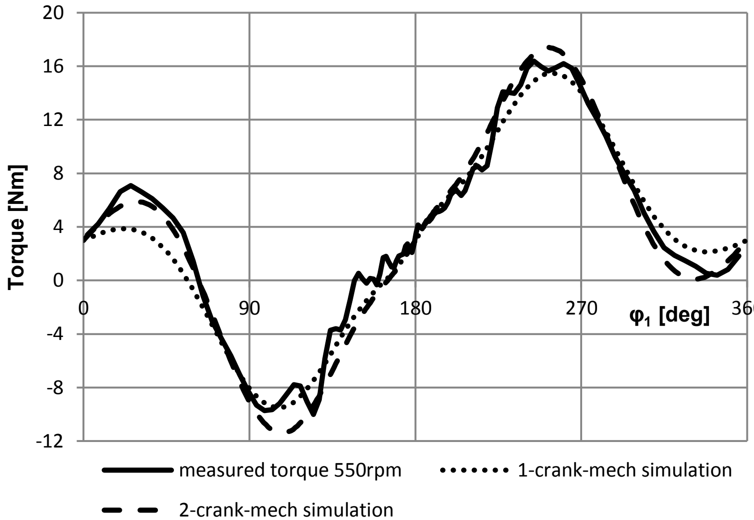

Fig. 5: Simulation/reality comparison – speed 550rpm

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.

For example the displacement law described by parameters κ and η like in (4.1) and (4.2) can be considered.

$$\varphi = \mu + \frac{\eta}{2\pi}\sin(2\pi\mu), \qquad (4.1)$$

where

$$\mu = \tau - \frac{\kappa}{2\pi}\frac{\sin(2\pi\tau)}{1+\kappa\cos(2\pi\tau)}, \quad (4.2)$$

It is possible to optimize the displacement law for every machine operating speed. It means to find the parameters κ and η and create the displacement law which satisfies the desired picking angle (it is different for each speed) and minimizes the computed driving torque. It can be very helpful because the optimal displacement law can increase the maximal speed of the machine. The standard optimization methods can be used.

5 Conclusions

The exact mathematical model of mechanical system is very important in order to predict the dynamic behavior of the particular mechanism. The suitable servomotor can be selected even the suitable displacement law can be computed for each operating speed. It is possible to find the optimal mechanical parameters too. The pre-computed drive torque can be used as a feed forward torque in the servomotor regulation which helps to reduce the position error and increase the quality of the weaving itself. It is also beneficial to have a simple mathematical model because the simulations are not too time consuming and then for instance the pre-computation of the feed forward torque can be even performed in the controller.

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