ANALYSIS OF DYNAMIC MODEL OF THE DRIVE OF SMALL DIAMETER KNITTING MACHINES ANGE 18.1

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Abstract

The paper deals with dynamic analysis of the driving system of small diameter knitting machines. Its main object is the description and analysis of the existing structural configuration of the drive. The present arrangement of the drive provides for a coupled motion of the needle cylinder and the dial, realized by a single driving unit. From the technological point of view, it is necessary to provide for a minimum deviation of swinging of principal parts of the machine during an operating cycle. The paper describes the compilation of a mathematical model. The motion equations have been devised by means of Lagrange equations of the second kind. They have been solved by means of the software Matlab and its superstructure Simulink. As the result, there have been obtained the courses of kinematic variables of basic parts of the driving system.

Introduction

The first step towards a modernization and optimization of the driving system is an evaluation of the existing state. The evaluation consists in an identification of courses of kinematic variables and determination of a maximum admissible deviation in the mutual swinging of the cylinder (*Fig. 1*, item V) and the dial (*Fig. 1*, item 8), which constitutes a technological condition. The cylinder and the dial are the two principal sub-systems of the knitting machine realizing the knitting process.

The *Fig. 1* shows a diagrammatical view of the drive, which represents the existing design of the driving system of small diameter knitting machines Ange, made by the manufacturer of small diameter knitting machines Uniplet Třebíč a.s.

The existing designing concept provides for a mechanical coupling between the motion of the needle cylinder (*Fig. 1*, item 9) and that of the dial of the knitting machine (8).

These two elements have equal r. p.m., and for a proper operation of the machine, their mutual adjustment must be very precise – given by a prescribed technological condition. This chain consists of gear wheels (*Fig. 1*, items 0-7) and shafts (*Fig. 1*, items 23, 45, 78).

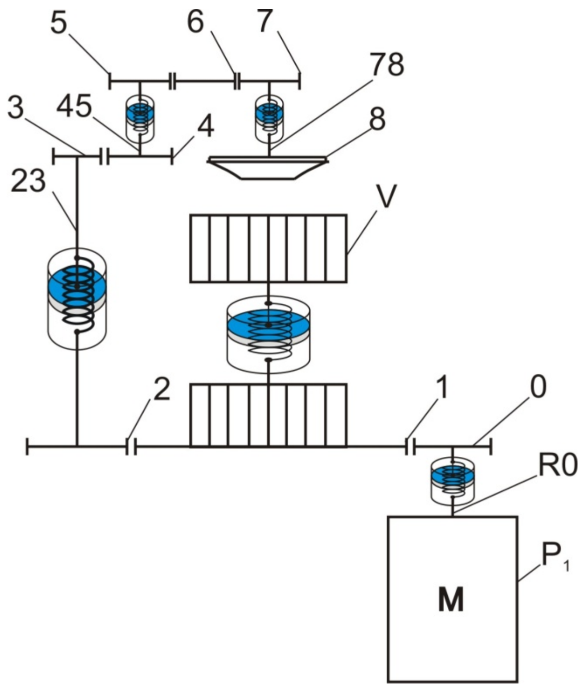

Fig. 1 Kinematic diagram of the drive of cylinder and dial of Ange 18.1

1 Mathematical model of the drive of Ange 18.1

In order to describe the behaviour of the studied system, it is necessary to devise a suitable mathematical model, able to describe its behaviour during an operating cycle with a defined precision. The individual parts of the system are influenced by a number of forces, the magnitudes of which are determined by the technological process and are variable in time. These forces include e.g. the forces necessary for lifting the needles and sinkers, the passive resistances in individual kinematic couples, passive resistances in the groove of the needle cylinder etc. However, from the view of the overall loading of the partial assemblies of the system, the magnitudes of certain forces are negligible, and in devising the mathematical model, they can be disregarded without an impact on the precision of the solution.

The proper mathematical model of the driving system of knitting machines (see *Fig. 1*) has been devised under the following conditions. The masses of individual elements of the mechanism (elements 0-8) including the pertinent parts of the shafts and of the seating are substituted by mass points. With the mass points, their inertia moments have been determined.

The shafts interconnecting the gear wheels transmitting the driving moment are considered to be elastic elements, and are substituted by torsional stiffness *k* (R0, 23, 45, 78). The mass of the needle cylinder 9 is considered including the needles, sinkers and its other components. It is divided into two mass points. These mass points are linked by a torsional spring. The elastic elements are attenuated by viscous damping with the damping co-efficient *b*.

1.1 Conditions

In the calculation there are considered the plays in gear wheels. These plays are substituted by angular deflections of gear wheels of the mechanism. The plays have been determined from drawings. For the compilation of motion equations of the system defined as above, there have been used the Lagrange equations of the second kind in the following form:

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c) the mass of the needle cylinder 9 is considered including the needles, sinkers and its other components. It is divided into two mass points. These mass points are linked by a torsional spring.
d) the elastic elements are attenuated by viscous damping with the damping co-efficient b.
e) in the calculation there are considered the plays in kinematic couples. These plays are substituted by angular deflections of gear wheels of the mechanism.

For the compilation of motion equations of the system defined as above, there have been used the Lagrange equations of the second kind in the following form:

$$\frac{d}{dt}\left(\frac{\partial E_K}{\partial \dot{q}_i}\right) - \frac{\partial E_K}{\partial q_i} + \frac{\partial E_P}{\partial q_i} = Q_j - \frac{\partial R_d}{\partial \dot{q}_i} \tag{1}$$

For the system in *Fig. 2*, we can elicit the following energy equations:

$$E_K = \frac{1}{2} I_0 \dot{\varphi}_0^2 + \frac{1}{2} I_1 \dot{\varphi}_1^2 + \frac{1}{2} I_V \dot{\varphi}_V^2 + \frac{1}{2} I_2 \dot{\varphi}_2^2 + \frac{1}{2} I_3 \dot{\varphi}_3^2 + \frac{1}{2} I_4 \dot{\varphi}_4^2 + \frac{1}{2} I_5 \dot{\varphi}_5^2 + \\ + \frac{1}{2} I_6 \dot{\varphi}_6^2 + \frac{1}{2} I_7 \dot{\varphi}_7^2 + \frac{1}{2} I_8 \dot{\varphi}_8^2 \tag{2}$$

$$E_P = \frac{1}{2} k_{R0} (\varphi_0 - \varphi_R)^2 + \frac{1}{2} k_V (\varphi_V - \varphi_1)^2 + \frac{1}{2} k_{23} (\varphi_3 - \varphi_2)^2 + \frac{1}{2} k_{45} (\varphi_5 - \varphi_4)^2 + \\ + \frac{1}{2} k_{78} (\varphi_8 - \varphi_7)^2 \tag{3}$$

$$R_d = \frac{1}{2} b_{R0} (\dot{\varphi}_0 - \dot{\varphi}_R)^2 + \frac{1}{2} b_V (\dot{\varphi}_V - \dot{\varphi}_1)^2 + \frac{1}{2} b_{23} (\dot{\varphi}_3 - \dot{\varphi}_2)^2 + \frac{1}{2} b_{45} (\dot{\varphi}_5 - \dot{\varphi}_4)^2 + \\ + \frac{1}{2} b_{78} (\dot{\varphi}_8 - \dot{\varphi}_7)^2 \tag{4}$$

The quantities appearing in the equations:

E_K- kinetic energy, E_P- potential energy, R_d- dissipative function, q_i –generalised co-ordinate, b_i - co-efficient of viscous damping [Nm.s.rad-1], k_i - co-efficient of stiffness [Nm.rad-1], - transmission, I -moment of inertia [kg.m2], φ- swinging [rad], $\dot{\varphi}$- angular velocity [rad.s-1], $\ddot{\varphi}$ - angular acceleration [rad.s^{-2}]

The subscripts in the designations of quantities express the relation to the elements of the drive. Once we perform the respective derivatives and insertions into the basic form of the Lagrange equation of the second kind, we obtain the actual motion equations. The number of equations corresponds to the respective generalised co-ordinates in the same sequence as the preceding derivative.

$$\ddot{\varphi}_0(I_0+I_1\eta_{01}^2+I_2\eta_{02}^2)+b_{0R}(\dot{\varphi}_0-\dot{\varphi}_R)-b_V(\dot{\varphi}_V-\dot{\varphi}_1)\eta_{01}-b_{23}(\dot{\varphi}_3-\dot{\varphi}_2)\eta_{02}+k_{0R}(\varphi_0-\varphi_R)- -k_V(\varphi_V-\varphi_1)\eta_{01}-k_{23}(\varphi_3-\varphi_2)=0 \quad (5)$$

$$I_V\ddot{\varphi}_V+b_V(\dot{\varphi}_V-\dot{\varphi}_1)+k_V(\varphi_V-\varphi_1)=0 \quad (6)$$

$$\ddot{\varphi}_3(I_3+I_4\eta_{34}^2)+b_{23}(\dot{\varphi}_3-\dot{\varphi}_2)-b_{45}(\dot{\varphi}_5-\dot{\varphi}_4)\eta_{34}+k_{23}(\varphi_3-\varphi_2)-k_{45}(\varphi_5-\varphi_4)\eta_{34}=0 \quad (7)$$

$$\ddot{\varphi}_5(I_5+I_6\eta_{56}^2+I_7\eta_{57}^2)+b_{45}(\dot{\varphi}_5-\dot{\varphi}_4)-b_{78}(\dot{\varphi}_5-\dot{\varphi}_4)\eta_{57}+k_{45}(\varphi_5-\varphi_4)- -k_{78}(\varphi_8-\varphi_7)\eta_{57}=0 \quad (8)$$

$$I_8\ddot{\varphi}_8+b_{78}(\dot{\varphi}_8-\dot{\varphi}_7)+k_{78}(\varphi_8-\varphi_7)=0 \quad (9)$$

The principal parameters for the solution of a dynamic model are the moments of inertia, stiffness and damping of individual elements of the drive of cylinder and dial of small diameter knitting machines. The principal parts of the driving system have been set up as a 3D model by means of the CAD software Pro/Engineer. By means of these models, there have been determined the inertia moments of the system elements, as well as the Method of final elements have been used to determine torsional stiffness of shafts and of needle cylinder. Another parameter included in the equations is the coefficient of viscous damping. The relation for calculation of the said coefficients is based on logarithmic decrement, moment of inertia and stiffness of the respective component. The plays have been included in the solution by means of the following conditions.

$$|\varphi_{iP}-\varphi_i|\le\Phi_i\Rightarrow\varphi_{iP}-\varphi_i=0 \quad (10)$$

$$\varphi_{iP}-\varphi_i>\Phi_i\Rightarrow\varphi_{iP}-\varphi_i\mapsto\varphi_{iP}-\varphi_i-\Phi_i \quad (11)$$

$$\varphi_{iP}-\varphi_i<-\Phi_i\Rightarrow\varphi_{iP}-\varphi_i\mapsto\varphi_{iP}-\varphi_i+\Phi_i \quad (12)$$

For i = 0, 1, 2, 3, 4, 5, 6, 7, 8.

Transmissions appearing in the equations

$\varphi_1=\varphi_0\eta_{01}$, $\varphi_2=\varphi_0\eta_{02}$, $\varphi_4=\varphi_3\eta_{34}$, $\varphi_6=\varphi_5\eta_{56}$, $\varphi_7=\varphi_5\eta_{57}$

The matrix shape of the movement equations

$$\mathbf{I}\ddot{q}+\mathbf{B}\dot{q}+\mathbf{K}q=0 \quad (13)$$

$$\ddot{\mathbf{q}} = -\mathbf{I}^{-1}(\mathbf{B}\,\dot{q} + \mathbf{K}\,q) \qquad (14)$$

For the derived motion equations there has been devised a mathematical model in the software Matlab. For the solution of these motion equations there has been employed the environment Matlab, including its superstructure Simulink. The individual equations have been written in matrix form, and subsequently solved by sequential integration by means of solver in the software Simulink. There has been employed a solver using the standard Dormand-Prince method, termed ode45.

By solution of the devised motion equations for the assigned initial conditions there has been obtained the response of the system to the prescribed kinematic excitation, namely to the course of acceleration during knitting the sock heel or toe. This knitting regime has been chosen because it makes highest demands on the driving system. The course of acceleration is idealised, and it is always assigned to the rotor of the electric motor.

Tab. 1 Mass and Materials properties

Component name	Part	Moment of inertia [Kg*m2]	Stiffness [N*m* rad^{-1}]	Damping [N*m*s*rad^{-1}]
gear wheel 0	0	9.8299361e-5	-	-
gear wheel 1	1	2.1271433e-2	-	-
gear wheel 2	2	1.3357134e-3	-	-
gear wheel 3	3	3.0747164e-5	-	-
gear wheel 4	4	2.8658272e-4	-	-
gear wheel 5	5	1.8609515e-4	-	-
gear wheel 6	6	2.5144821e-4	-	-
gear wheel 7	7	1.49e-4	-	-
Dial	8	2.4e-4	-	-
Needle cylinder	V	2.3864947e-2	200750	25.5625
Shaft R0	R0	8.1e-4	12954	1.4087
Shaft 23	23	72.6e-6	3423	0.1100
Shaft 45	45	8.425e-5	4500	0.1359
Shaft 78	78	2.96e-6	2560	0.0192

2 Results of the dynamic analysis

The results of the analysis indicate an important effect of plays in the system. They bring about high impact forces at a change of direction of the movement of individual gear wheels in gear sets. In the *Fig. 4* there can be seen considerable peaks in acceleration, due namely to the adjustment of plays and the subsequent mutual impact of individual dents.

These values are as much as 3 times larger in comparison with a system without plays. In a similar way – although not so considerably –the adjustment of plays manifests itself in the courses of the velocity (*Fig. 5*), too. A deformation of the course of velocity in comparison with the theoretical course is notable here as well. The *Fig. 3* displays the difference of velocities of the needle cylinder and of the dial. Plays in the system and the resiliency of individual elements of the system bring about deviations in the position of the output element of the kinematic structure (dial – element 8). The course of the difference in the swinging of the dial with respect to the needle cylinder is shown in the *Fig. 2*.

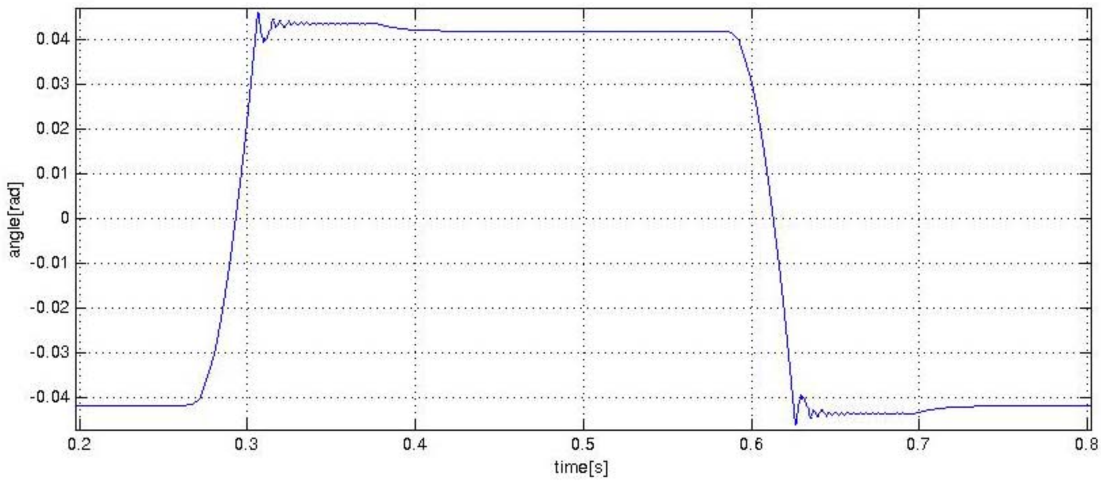

Fig. 2 Difference of positions of the cylinder and the dial

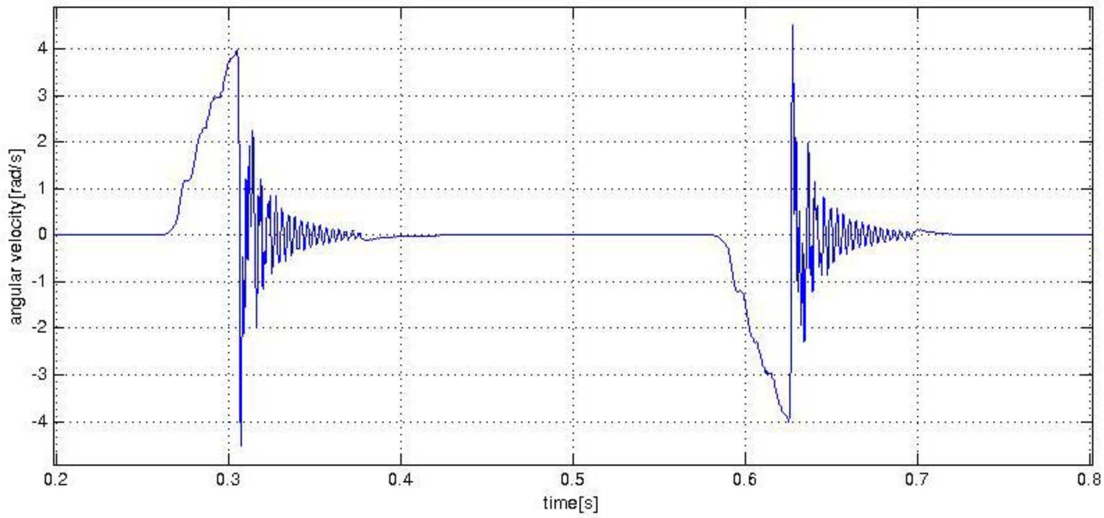

Fig. 3 Difference of velocities of the cylinder and the dial

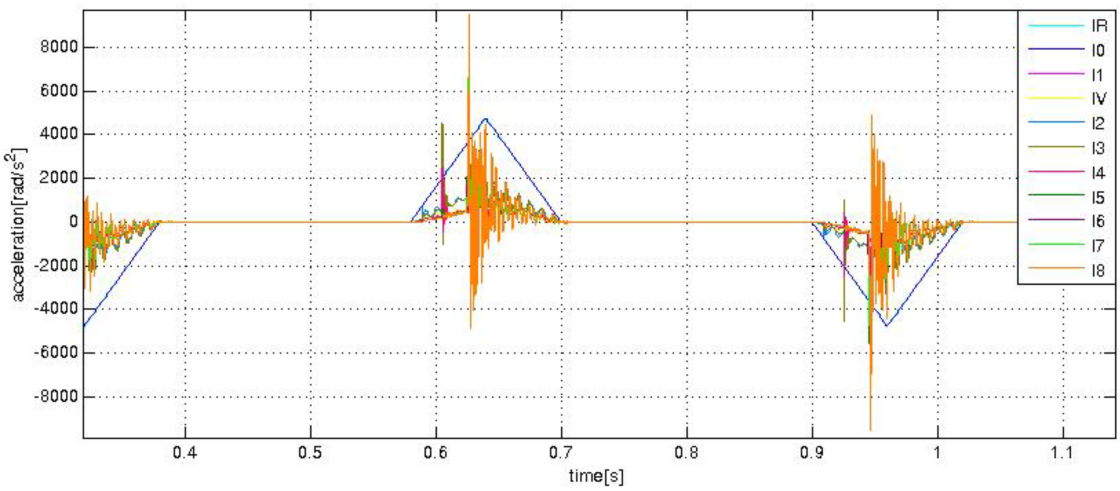

Fig. 4 Courses of accelerations of individual elements of the drive

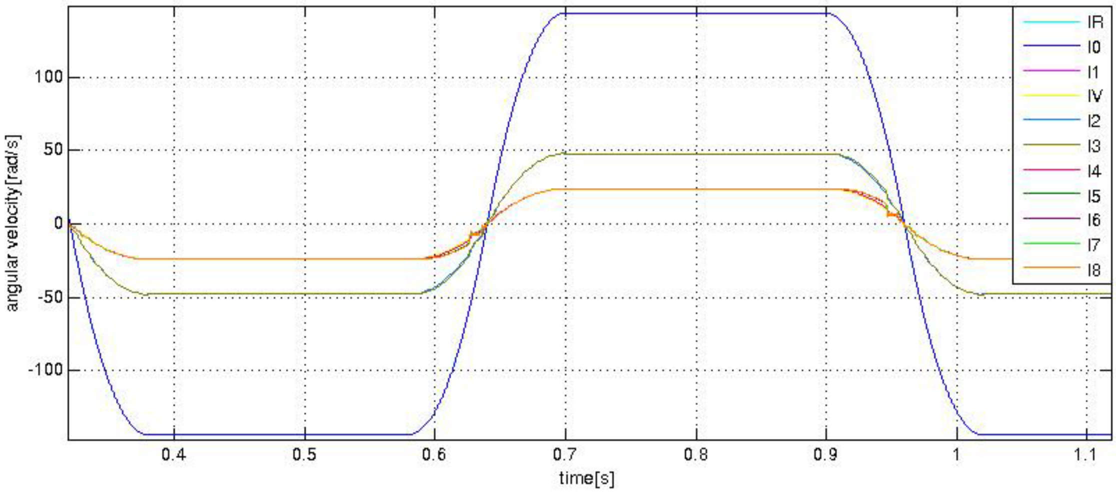

Fig. 5 Courses of velocities of individual elements of the drive

Conclusion

The paper describes the compilation of a mathematical model of the drive of small diameter knitting machines. As the result, there have been obtained the courses of kinematic quantities, and there has been determined the maximum deviation of the positions of the cylinder and of the dial in the worst loading condition. There has been studied the system of knitting machines ANGE 18.1, where the analysis has brought the value of the maximum deviation of positions 4.8 °.

A previous study [1] has demonstrated the advantages of a reduction of the number of elements in the driving system. This structural modification reduces the energy required for the drive of the principal parts of the machine by as much as 30%. It can be achieved by employment of controlled drives. Based on these facts, there will be devised a new mathematical model, taking in consideration a new configuration of the driving system with the application of unit drives.

The compiled mathematical model will serve as optimization tool for subsequent dynamic tuning of the system. Employing this model, it will be possible to design driving systems and to analyse varied operating regimes of the driving unit.

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ANALÝZA DYNAMICKÉHO MODELU POHONU MOLOPRŮMĚROVÉHO PLETACÍHO STROJE ANGE 18.1

Článek je zaměřen na dynamickou analýzu pohonného systému maloprůměrového pletacího stroje. Hlavním cílem je popis a analýza stávajícího konstrukčního uspořádání náhonu. Současné uspořádání pohonu zajišťuje svázaný pohyb jehelního válce a přístroje realizovaný jednou pohonnou jednotkou. Z technologického hlediska je nutné zajistit minimální odchylku natočení hlavních částí stroje během pracovního cyklu. Příspěvek popisuje sestavení matematického modelu. K sestavení pohybových rovnic byly použity Lagrangeovy rovnice druhého druhu. Ty byly řešeny pomocí software Matlab a jeho nadstavby Simulink. Výsledkem jsou průběhy kinematických veličin základních částí pohonného systému.

ANALYSE DES DYNAMISCHEN MODELLS DES ANTRIEBS DER KLEINRUNDSTRICKMASCHINE ANGE 18.1

Der Artikel befasst sich mit der dynamischen Analyse des Antriebssystems der Kleinrundstrickmaschine. Das Hauptziel ist die Beschreibung und die Analyse der bestehenden Konstruktionsgestaltung des Antriebs. Die derzeitige Anordnung des Antriebs gewährleistet die gekoppelte Bewegung des Nadelzylinders und des Gerätes mittels einer Antriebseinheit. Unter dem technologischen Aspekt ist es erforderlich, eine minimale Abweichung der Drehung der Hauptteile während des Arbeitszyklus zu gewährleisten. Der Beitrag beschreibt die Erstellung eines mathematischen Modells. Zur Formulierung der Bewegungsgleichungen wurden Lagrangegleichungen zweiter Art verwendet. Diese wurden mittels der Software Matlab und ihres Überbaus Simulink gelöst. Das Ergebnis sind die Verläufe der kinematischen Größen der grundlegenden Teile des Antriebssystems.

ANALIZA MODELU DYNAMICZNEGO NAPĘDU MAŁOŚREDNICOWEJ MASZYNY DZIEWIARSKIEJ ANGE 18.1

Artykuł jest poświęcony analizie dynamicznej układu napędowego małośrednicowej maszyny dziewiarskiej. Głównym celem jest opis i analiza obecnego układu konstrukcyjnego napędu. Obecnie układ napędu zapewnia sprzężony ruch cylindra igłowego i przyrządu wykonywany przez jedną jednostkę napędową. Z technologicznego punktu widzenia konieczne jest zapewnienie minimalnego odchylenia ustawienia głównych części maszyny w ciągu cyklu roboczego. Artykuł opisuje zestawienie modelu matematycznego. Do zestawienia równań ruchu wykorzystano równania Lagrange’a drugiego rzędu. Rozwiązywano je przy pomocy oprogramowania Matlab i jego nadbudowy Simulink. Wynikiem są przebiegi wielkości kinematycznych podstawowych części układu napędowego.