THE DYNAMIC ANALYSIS OF THE NEEDLE BAR MECHANISM OF SEWING MACHINES

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Abstract

The article is concerned with the dynamic analysis of the mechanism of the needle transfer, which has been carried out by means of Lagrange equations of the second kind, with the purpose to determine kinematic magnitudes of the individual elements. There have been established geometrical and physical properties of the individual elements, and the initial conditions have been determined. Furthermore, there have been compiled motion equations, and they have been complemented with the boundary conditions providing for the proper function of the mechanism. A program for the solution of the proper motion equations has been generated in the environment of Matlab Simulink, including the boundary and initial conditions. The results of the dynamic analysis have been processed graphically.

Introduction

The present trend in the development of sewing machines is to shorten the sewing times in the sewing process and to increase the productivity. The endeavour to increase the productivity of these machines requires a thorough understanding of all processes related to the operation of sewing machines. The object of this study is to perform an identification of the needle transfer mechanism, which will lead to the determination of kinematic magnitudes of the individual elements of this mechanism. The identification of the mechanism of the needle transfer will be carried out by the analytic method of Lagrange equations of the second kind, which allow formulating the laws of motion by means of scalar quantities. The proper solution of the motion equations obtained by Lagrange method will be carried out by means of the software Matlab Simulink.

1 Description of the needle transfer mechanism

The needle transfer mechanism performs a rectilinear reverse movement that has been realised by a cam mechanism in the existing machine [2]. The newly proposed functional model of the machine consists of a crank mechanism with servo-drive converting rotary swinging motion into the rectilinear reverse movement. The needle transfer mechanism (*Fig. 1*) forms a part of the system that allows imitating the hand stitch. Its task consists in providing for the transfer of the floating needle between two needle bars operating above the work table of the machine and below it. The floating needle is clamped by collets (*Fig. 1,* item 12) in the needle bar, owing to the pressing force of the springs, items 11 and 15. The unlocking of the collets is actuated during the movement of the needle bar by the impact of its controlling element (items 1, 6, 11, 3, 10, 17, 16) upon the machine frame before the dead centre of the needle transfer, where the impact is damped by a rubber pad (item 10). The stop block can be seen in *Fig. 2*.

The jacket of the needle bar (*Fig. 1,* item 2) goes on moving to the bottom dead centre and completes the process of the needle transfer.

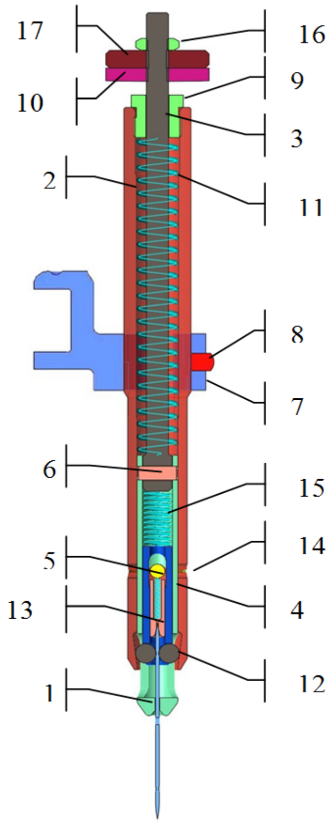

Fig. 1 *Sectional view of the needle transfer mechanism*

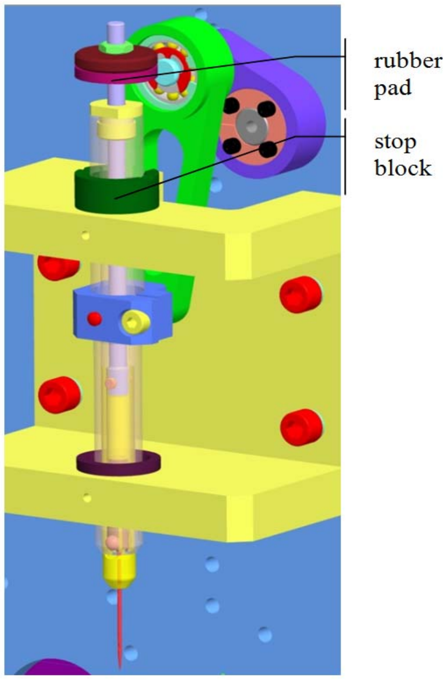

Fig. 2 *Newly proposed crank mechanism*

There has been carried out experimental measuring of the force response of the springs (see *Fig. 1*, items 11 and 15) in order to establish the stiffness of these springs [1], and there has been taken a record by a high-speed camera, leading to determine the values of their intrinsic damping [1]. The results of these measurements are shown in *Tab. 1*. Another flexible element is the rubber pad (*Fig. 2*) mentioned previously. As for this pad, there has been carried out experimental measuring in order to establish the force required for the compression of the rubber pad. The resulting diagram of this force response is shown in the *Fig. 3*. In the record there can be seen the expected non-linear behaviour of the rubber pad during its compression. The course of the acting force has been approximated by a polynomial of the third degree (1) where x stands for the deformation of the rubber pad [1]. The equation of the dissipative force of the rubber pad is given by the relation (2). The coefficient of the linear damping figuring in the equation (2) is determined from the real behaviour of the system of the needle mechanism [1]. The mechanism of the needle bar and its proper function are provided by the system of the stop blocks. The coefficient of the restitution of these stops has been adjusted to such a value that the behaviour of the mathematical model would be as close as possible to the behaviour of the real mechanism which has been examined by means of the high-speed camera.

Tab. 1 Values of stiffness and intrinsic damping of the springs

	Stiffness k_j [N/m]	Intrinsic damping b_j [N.s/m]
Spring item 11 (index 1)	900	13.16
Spring item 15 (index 2)	690	3.32
Rubber pad item 10 (index 3)	-	28.5

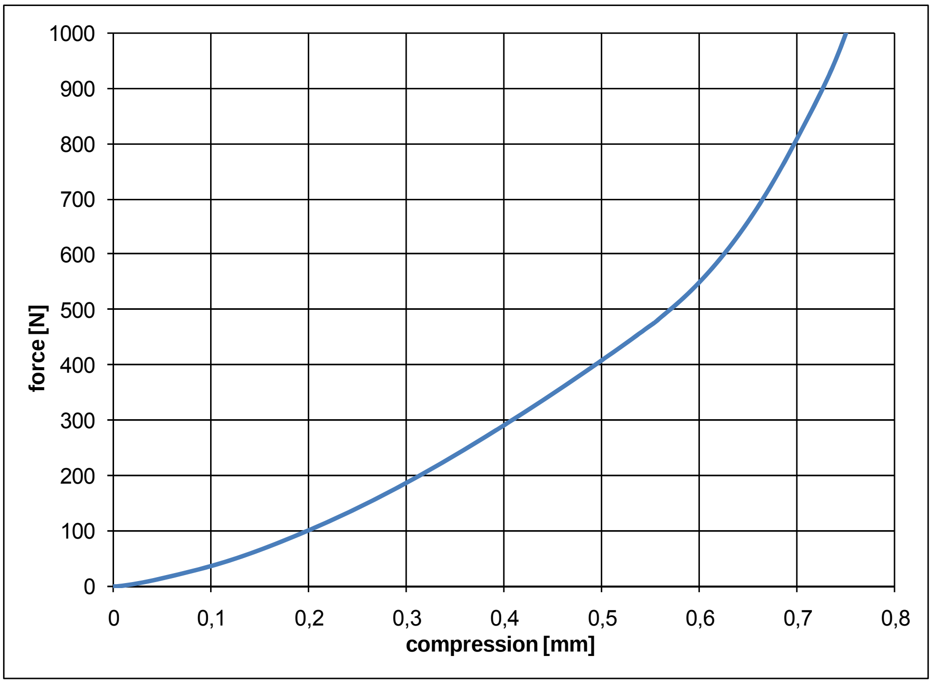

Fig. 3 Force necessary for compression of the rubber pad

$$F_3(x) = 1573{,}9x^3 - 2e^{-10}\,x^2 + 445{,}17x \quad (1)$$

$$F_4 = b_3\dot{x}_2 \quad (2)$$

2 Description of the mathematical model

The mechanism of the needle transfer has been simplified to three elements 1, 2 and 3 only, of the masses m_1, m_2 and m_3 (see *Fig. 5*). The element 1 represents the jacket of the needle bar mechanism, the element 2 represents the controlling element of the mechanism and the element 3 represents the collets with the cage gripping the needle. The individual elements of the mechanism are represented by the perfectly rigid bodies and they are interconnected by springs and dampers. The element 1 is excited kinetically. The course of the excitement is shown in the following *Fig. 4* and it matches the machine speed at 250 cpm. Moreover, the element 2 is influenced by the forces F_3 and F_4. The force F_3 arises from the effect of the rubber pad (see *Fig. 1.* and *Fig. 2*) and the force F_4 is the dissipative force. These two forces bear upon the body only in the moment when the position of the element 2 reaches the position of the stop (see *Fig. 2*).

The analysis of the needle transfer mechanism has been carried out for two variants. The first variant has been the model of the original concept, and the other one the model of the optimised concept with a modified inner cylinder. *Tab. 2* presents the initial conditions for the simplified mathematical model. *Tab. 3* resumes geometrical values of the model including other parameters necessary for the description of the system.

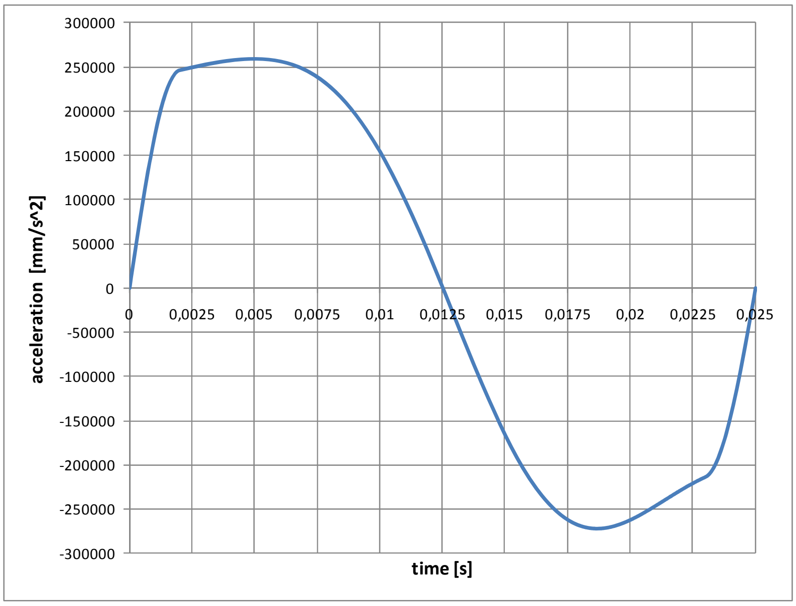

Fig. 4 *The course of acceleration of the element 1 during the movement from the upper position to the lower one*

Tab. 2 *Initial conditions for the needle bar mechanisms of original and optimised designs*

Mech. of needle bar – orig. design	Element 1	Element 2	Element 3
Initial position [m]	0	0.0719	0.0929
Initial value of velocity [m/s]	0	0	0
Initial value of acceleration [m/s^2]	0	0	0

Mech. of needle bar – opt. design	Element 1	Element 2	Element 3
Initial position [m]	0	0.0669	0.0929
Initial value of velocity [m/s]	0	0	0
Initial value of acceleration [m/s^2]	0	0	0

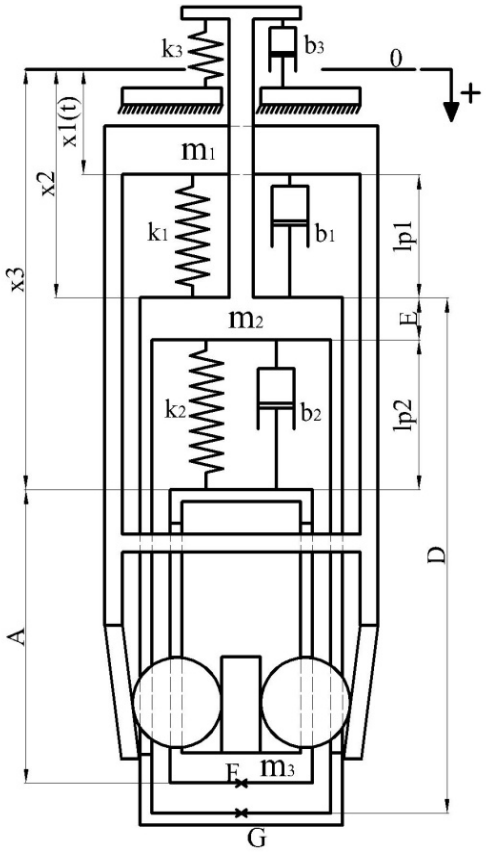

Fig. 5 Scheme of simplified mode of needle bar mechanism

Tab. 3 Geometrical and mass magnitudes of system elements

Mass m_1 of element 1[kg]	0.1196
Mass m_2 of element 2 [kg]	0.0426
Mass m_3 of element 3 [kg]	0.0051
Unmount.length of spring l_{01} [m]	0.089
Unmount.length of spring l_{02} [m]	0.0292
Value A [m]	0.0268
Value D [m]	0.0547
Value E [m]	0.0081
Length of spring after mounting l_{p1} [m] in original design	0.0718
Length of spring after mounting l_{p2} [m] in original design	0.0129
Length of spring after mounting l_{p1} [m] in optimised design	0.0668
Length of spring after mounting l_{p2} [m] in original design	0.0179

3 Compilation of motion equations

In order to obtain the motion equations, there has been employed the analytic method of Lagrange equations of the second kind for the homonomous couplings. The equation (3) has the usual form of Lagrange equations of the second kind (LEIID) for the system with potential, dissipative and operating forces, where q_j stands for the generalised coordinate, E_k for the kinetic energy of the system, E_p for the potential energy of the system, R_d for the dissipative energy and Q_j for the operating force [4]. In our case, the generalised coordinates are x_2 and x_3. The values of the intrinsic damping b_j and the stiffness of the springs k_j have been taken from *Tab. 1*. This mathematical model does not include the force of gravity.

$$\frac{d}{dt}\left(\frac{\partial E_K}{\partial \dot{q}_j}\right)-\frac{\partial E_K}{\partial q_j}=Q_j-\frac{\partial E_P}{\partial q_j}-\frac{\partial R_d}{\partial \dot{q}_j} \quad . \tag{3}$$

First of all, for the chosen analytic method LEIID the kinetic energy of the whole system (4) has been determined.

$$E_k=\frac{1}{2}m_2\dot{x}_2^2+\frac{1}{2}m_3\dot{x}_3^2. \tag{4}$$

Next, the potential energy of the whole system (5) has been established.

$$E_P=\frac{1}{2}k_1(x_2-x_1-l_{01})^2+\frac{1}{2}k_2(x_3-x_2-l_{02}-E)^2+\left(\frac{1573{,}9}{4}(x_2-0{,}0949)^4-\frac{2\times10^{-10}}{3}(x_2-0{,}0949)^3+\frac{445{,}17}{2}(x_2-0{,}0949)^2\right). \tag{5}$$

The dissipative energy of the whole system is given by the sum of the individual components of the dissipative function (6).

$$R_d=\frac{1}{2}b_1(\dot{x}_2-\dot{x}_1)^2+\frac{1}{2}b_2(\dot{x}_3-\dot{x}_2)^2+\frac{1}{2}b_3\dot{x}_2^2 \quad . \tag{6}$$

The equation (3) shows the component of the operating force. In the system of the needle transfer mechanism there is no function of any operating force component; therefore, the element Q_j=0. In the next step, the derivatives of the individual energy components according to the respective coordinates have been found. The derivative of kinetic energy according to the individual coordinates is

$$\frac{d}{dt}\left(\frac{\partial E_K}{\partial \dot{x}_2}\right)=m_2\ddot{x}_2 \tag{7}$$

and

$$\frac{d}{dt}\left(\frac{\partial E_K}{\partial \dot{x}_3}\right)=m_3\ddot{x}_3 \, . \tag{8}$$

The derivative of the potential energy according to the individual coordinates (x_2, x_3) is

$$\frac{\partial E_K}{\partial x_2}=k_1(x_2-x_1-l_{01})-k_2(x_3-x_2-l_{02}-E)+[1573{,}9(x_2-0{,}0949)^3-2\times10^{-10}(x_2-0{,}0949)^2+445{,}17(x_2-0{,}0949)] \tag{9}$$

and

$$\frac{\partial E_K}{\partial x_3}=k_2(x_3-x_2-l_{02}-E) \quad . \tag{10}$$

The derivative of the dissipative energy indicated by the sign is

$$\frac{\partial R_d}{\partial \dot{x}_2} = b_1(\dot{x}_2 - \dot{x}_1) - b_2(\dot{x}_3 - \dot{x}_2) + b_3\dot{x}_2 \tag{11}$$

and

$$\frac{\partial R_d}{\partial \dot{x}_3} = b_2(\dot{x}_3 - \dot{x}_2)\,. \tag{12}$$

Subsequently, it is possible to perform the substitutions in the equation (3) and to compile the motion equations for the elements 2 and 3 of the system. The relation (13) constitutes the motion equation for the element 2 and the relation (14) is the motion equation for the element 3 of the system. It is

$$m_2\ddot{x}_2 = -k_1(x_2 - x_1 - l_{01}) + k_2(x_3 - x_2 - l_{02} - E) - \{1573{,}9(x_2 - 0{,}0949)^3 - 2 \times 10^{-10}(x_2 - 0{,}0949)^2 + 445{,}17(x_2 - 0{,}0949)\} - b_1(\dot{x}_2 - \dot{x}_1) + b_2(\dot{x}_3 - \dot{x}_2) - b_3\dot{x}_2 \tag{13}$$

and

$$m_3\ddot{x}_3 = -k_2(x_3 - x_2 - l_{02} - E) - b_2(\dot{x}_3 - \dot{x}_2) \tag{14}$$

For the generation of the mathematical model there have been determined the boundary conditions simulating the system of stop blocks and cams which provide for the proper functioning of the needle transfer mechanism in the real model.

$$If\ x_2 \geq x_1 + l_{p1}\ \ then\ \ddot{x}_2 = \ddot{x}_1 \tag{15}$$

$$If\ x_3 \geq x_2 + l_{p2} + E\ \ then\ \ddot{x}_3 = \ddot{x}_1 \tag{16}$$

$$If\ x_2 \geq 0{,}0989\ \ then\ \ddot{x}_2 \neq \ddot{x}_1 \tag{17}$$

$$If\ x_2 \geq 0{,}0989\ \ then\ F_3, F_4 \neq 0 \tag{18}$$

$$If\ x_2 + D = x_3 + A\ \ then\ \ddot{x}_3 \neq \ddot{x}_1 \tag{19}$$

$$If\ x_2 + D = x_3 + A\ \ and\ \dot{x}_3 = 0\ then\ \ddot{x}_3 = \ddot{x}_2\,, \dot{x}_3 = \dot{x}_2 \tag{20}$$

The boundary condition (15) simulates the impact of the element 2 upon the transversal of the pin guided by the element 1. The condition (16) simulates the stop block consisting of two balls resting on the conical surface of the element 1. The conditions (17) and (18) simulate the impact of the rubber pad (or the element 2) on the machine frame. This impact comes up as soon as the element 2 reaches the position 0.0989 [m]. In this moment, the forces F_3 and F_4 start to act. The conditions (19) and (20) indicate that in the moment of the contact of the points F and G (see *Fig. 3*) the element 3 impacts with the element 2. The coefficient of the restitution between the elements 2 and 3 has been set up to the value 0.6. In the moment when the element 3 does not bounce from the element 2 anymore, the condition (20) is fulfilled. In this moment, the kinematic magnitudes (acceleration and velocity) of the element 3 are identical with the values of the acceleration and velocity of the element 2. The elements 2 and 3 perform a rectilinear reverse movement still, caused by the effect of the force by the rubber pad.

The equations (13, 14) have been resolved by means of the software Matlab with Simulink module.

4 Results of the kinematic analysis

The results of the kinematic magnitudes (position, velocity, acceleration) are shown in the following figures. *Figs 6-8* show the courses of the kinematic magnitudes of the elements 1, 2, 3 for the original concept and for the optimised concept. *Fig. 9* shows the courses of the positions of the points F and G for the original concept and for the optimised concept.

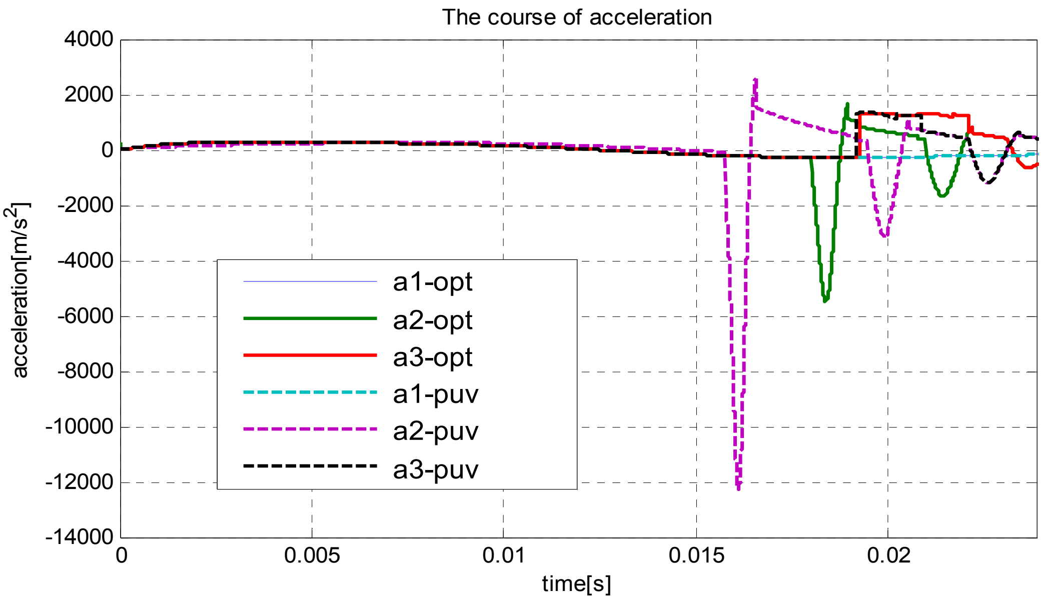

Fig. 6 Acceleration of elements 1, 2, 3 for the mechanisms of original design (puv) and of the optimised one (opt)

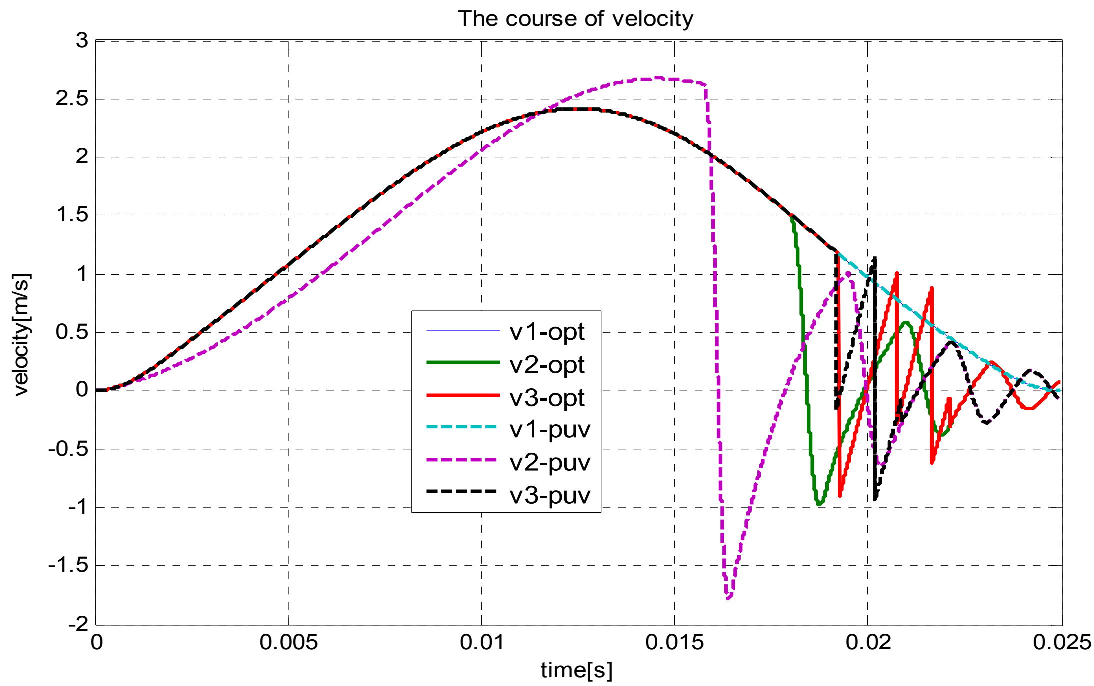

Fig. 7 Velocity of elements 1, 2, 3 for the mechanisms of original design (puv) and of the optimised one (opt)

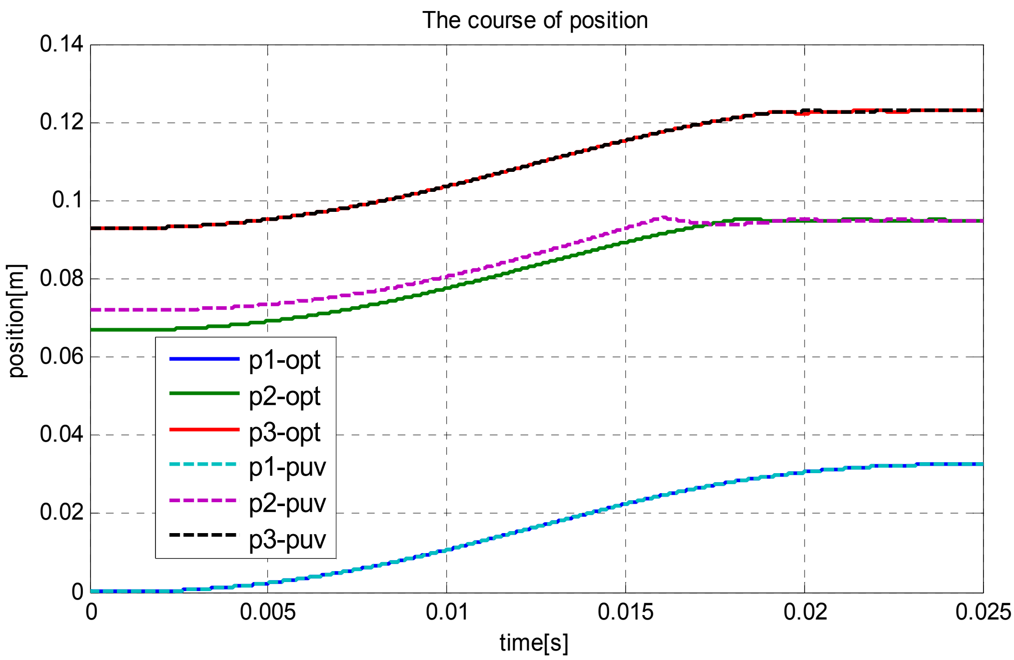

Fig. 8 *Position of elements 1, 2, 3 for the mechanisms of original design (puv) and of the optimised one (opt)*

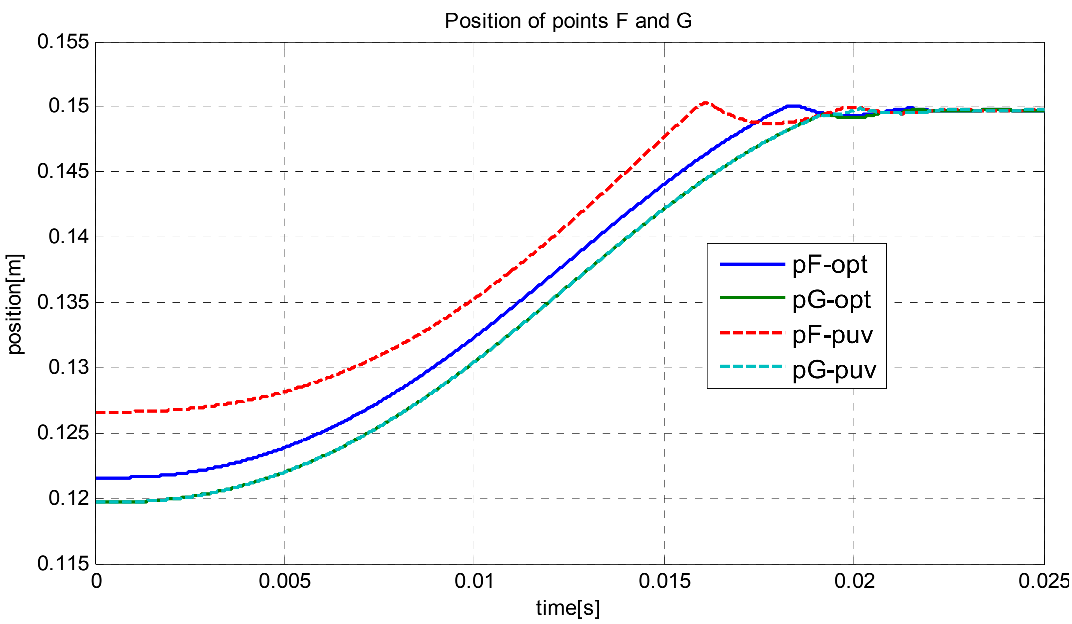

Fig. 9 *Positions of the points F and G for the mechanisms of original design (puv) and of the optimised one (opt)*

Fig. 6 to *Fig. 8* show that - in the original concept of the needle transfer mechanism - the controlling element is disengaged from the jacket in the machine speed at 250 cpm. Owing to this disengagement of the controlling element, the velocity of the impact of the element 2 on the stop block increases as well – see *Fig. 4*. In the original concept, at the time t=0.01575s the rubber pad is compressed and the element 2 bounces from the stop block. In the optimised concept, the rubber pad gets compressed at the time t=0.01796s. In the original design, the maximum acceleration of the element 2 comes up at the time t=0.0161s meanwhile in the optimised design the maximum acceleration comes up at the time t=0.01815s. At the time t=0.01923s, the elements 3 and 2 get in mutual contact (see *Fig. 9*). The element 1 goes on completing its movement up to the value 0.0326m, which is necessary for the sufficient opening of the collets.

Conclusion

There has been generated a simplified model representing the mechanism of the needle bar, and the geometrical and mass parameters of the system elements have been determined. Subsequently, the motion equations have been compiled by means of Lagrange equations of the second kind. These motion equations have been complemented with the initial and boundary conditions. The motion equations have been solved in the environment of Matlab Simulink and the results have been processed graphically.

It has been ascertained that during the movement from the upper position of the needle transfer mechanism to the lower there are produced the impacts generating the high values of the acceleration of the element 2. Moreover, there occur the mutual impacts of the elements 2 and 3. These impacts are the source of the high levels of the noise and vibration intensity transferred upon the machine frame, which is confirmed by the experimental measuring performed on the real machine, too [5]. This mathematical model of the needle bar mechanism will constitute a part of the total identification of the system in the future, too, together with the crank mechanism and the servo-drive.

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Literature

[1] PEJCHAR, K., BERAN, J.: Optimization of the needle bar mechanism /article/. In. *X. International Scientific Conference Transfer 2009*, 17th – 18th November 2009. Trenčín, p. 44. Slovak Republic. ISBN 978—80—8075—414—3

[2] PEJCHAR, K.: *Analysis and optimization of dynamic behavior of sewing machines*. /Diploma work/. Liberec, TUL 2008,

[3] BERAN, J., PEJCHAR, K., KOMÁREK, J.,: Analysis and optimization of needle bar transfer mechanism. *ACC JOURNAL 2010*, year XVI, Issue A, Natural Sciences and Technology, ISSN 1803-9782

[4] JULIŠ, K., BREPTA, R.,: *Mechanika II Dynamika*, Volume 1, SNTL Praha 1987, L13-E1-V-41f/2270

[5] PEJCHAR, K., BERAN, J., Experimental analysis of the mechanism of sewing machine. *Acta Universitatis Cibiniensis*, Vol. LVIII, 2009, s. 55 - 60, Sibiu, Romania, ISSN 1583 - 7149.

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DYNAMICKÁ ANALÝZA MECHANISMU JEHELNÍ TYČE U ŠICÍCH STROJŮ

Článek pojednává o dynamické analýze mechanismu předávání jehly, která byla provedena pomocí Lagrangeových rovnic druhého druhu s cílem určit kinematické veličiny jednotlivých členů. Byly určeny geometrické a fyzikální vlastnosti jednotlivých členů a určeny počáteční podmínky. Dále byly sestaveny pohybové rovnice, které byly doplněny o okrajové podmínky zajišťujícími správnou funkci mechanismu. V prostředí Matlab Simulink byl vytvořen program pro řešení vlastních pohybových rovnic včetně okrajových a počátečních podmínek. Výsledky dynamické analýzy byly graficky zpracovány.

DYNAMISCHE ANALYSE DES MECHANISMUS DER NADELSTANGE BEI NÄHMASCHINEN

Der Artikel behandelt die dynamische Analyse des Mechanismus des Übergebens der Nadel, die mittels der Lagrange-Gleichungen zweiter Art mit dem Ziel erfolgte, die kinematischen Größen der einzelnen Glieder zu bestimmen. Es wurden die geometrischen und physikalischen Eigenschaften der einzelnen Glieder sowie die Anfangsbedingungen bestimmt. Ferner wurden Bewegungsgleichungen aufgestellt, die um die Randbedingungen ergänzt wurden, die die korrekte Funktion des Mechanismus gewährleisten. Im Medium Matlab Simulink wurde ein Programm für die Lösung der eigenen Bewegungsgleichungen, einschließlich der Rand- und Anfangsbedingungen, erstellt. Die Ergebnisse der dynamischen Analyse wurden grafisch dargestellt.

ANALIZA DYNAMICZNA MECHANIZMU PROWADNICY IGŁY W MASZYNACH SZWALNICZYCH

Artykuł traktuje o analizie dynamicznej mechanizmu przemieszczania igły, którą przeprowadzono z pomocą równań Lagrange‘a drugiego rzędu, w celu określenia wielkości kinematycznych poszczególnych elementów. Zostały określone właściwości geometryczne i fizyczne poszczególnych elementów oraz określono warunki początkowe. Następnie zestawiono równania ruchu, które dopełniono warunkami brzegowymi zapewniającymi prawidłowe funkcjonowanie mechanizmu. W środowisku Matlab Simulink opracowano program do rozwiązywania właściwych równań ruchu wraz z warunkami brzegowymi i początkowymi. Wyniki analizy dynamicznej opracowano w formie graficznej.