Multiobjective Constrained Optimization of a Newly Developed Needle Driving Mechanism in Sewing Machine for Performance Improvement

S. Ebrahimi*
Department of Mechanical Engineering,
Yazd University, Iran
E-mail: ebrahimi@yazd.ac.ir
*Corresponding author

I. Hajizadeh & P. Payvandy
Department of Mechanical Engineering,
Yazd University, Iran
E-mail: hajizaqdeh.iaman@stu.yazd.ac.ir, peivandi@yazd.ac.ir

Received: 28 April 2014, Revised: 21 June 2014, Accepted: 30 June 2014

Abstract: Sewing is one of the most commonly used manufacturing processes in the world. In the textile industry, development of sewing machines with optimal mechanical performance is very important. Obviously, the quality of sewing, increase of the needle transmission force and the optimal mechanical advantage are greatly dependent on the design criteria of the needle driving mechanism. Therefore, in this paper, first a newly developed needle driving mechanism of a sewing machine is introduced. Then, the concepts of transmission angle and mechanical advantage are described. Next, the multiobjective constrained optimization of this mechanism using the genetic algorithm is explained. The objective functions of the optimization problem are considered in such a way to fulfil some of the most important design criteria such as reducing the needle generated heat, reducing undesirable vibrations and increasing the mechanical performance. The obtained results confirm improvement of the required design criteria of the newly developed mechanism in this study. It is also concluded that improving the mechanical advantage of about 14% causes an increase in the value of the needle jerk about 30%. This clearly states that higher mechanical advantage, is achieved by the cost of increasing the needle jerk.

Keywords: Genetic Algorithm, Optimization, Sewing Machine, Transmission angle

Reference: Ebrahimi S., Hajizadeh, I., and Payvandy, P., "Multiobjective Constrained Optimization of a Newly Developed Needle Driving Mechanism in the Sewing Machine for Improvement of the Mechanical Performance", Int J of Advanced Design and Manufacturing Technology, Vol. 7/ No. 3, 2014, pp. 9-18.

Biographical notes: S. Ebrahimi received his PhD in Mechanical Engineering from Stuttgart University, Germany in 2007. He is currently Assistant Professor at the Department of Mechanical Engineering, Yazd University, Yazd, Iran. His current research interest includes dynamic modelling of Multibody Systems and Robotics. **P. Payvandy** is Assistant Professor of Textile Engineering at the Yazd University, Iran. He received his PhD in Textile Engineering from Amirkabir University of Iran. His current research focuses on optimization of sewing machines.

1 INTRODUCTION

Textile industry and its applications to many industrial categories has found great attention in recent years. Nowadays, in the realization of the required level of quantity and quality of textile products, the application of several engineering fields in textile industry has shown considerable influence. In this context, development of new sewing machines with high level of mechanical performance is of great importance. In recent years, many researchers have focused on development and application of optimization algorithms in engineering design problems. However, there is still a clear gap in implementing these approaches in synthesizing mechanisms of sewing machines. As few examples in this context, vibrations reduction of industrial sewing machines by eccentric balancing of crank mechanism of the needle bar is presented in [1], [2].

The authors of deal with method of balancing optimization of a very complex set of sewing machine mechanisms using Adams and Mathematica Softwares [3]. The work presented by Komarec et al., is concerned with an analysis and optimization of the needle transfer mechanism by means of the software Pro/Engineer Wildfire 4 with the aim to obtain a reduction of needle wear [4]. In textile engineering, sewing machines have been basically designed for work at high velocity. One of the most important problems in the process of sewing, is the heat generated by friction between fabric and needle in the penetration zone [5]. The heat causes severe problems during the sewing process, such as thread and fabric melting as well as residues around the needle's eye.

One of the effective approaches for reduction of the needle temperature is based on the modification of the needle velocity of the needle driving mechanism in the penetration zone. According to literature, few researches have focused to design a needle driving mechanism of the sewing machine to reduce the generated needle heat. As an important study in this area, the possibility of replacing the typical slider-crank mechanism with a link drive mechanism was proposed in [6]. Further modification of this mechanism was then carried out by implementing an optimization algorithm using the imperialistic competitive algorithm in [7]. However, reduction of the needle velocity in the penetration zone may affect the amount of required needle force which in turn depends on the transmission angle and mechanical advantage of the sewing machine [8]. Deviation of the transmission angle from 90° is a measure of reduction in effectiveness of the force transmission. The major goal in this context is to find a configuration for which the transmission angle experiences low variation from its ideal value.

As one of the earliest works in this context, Shoup and Pelan have developed a technique to simultaneously optimize transmission angle and structural error [9]. Graphical techniques presented by Sun and Waldron permit to control the maximum of the transmission angles in the design positions [10]. Soylu has investigated the transmission angle synthesis of single degree-of-freedom mechanism [11]. The method is applicable to any spatial or planar mechanism for which a transmission angle may be defined. Balii and Chand have proposed importance of the transmission angle for the most effective force transmission [12]. Tsai and Lai have considered an effective method to analyze the transmission performance of linkages with joint clearances [13]. Yildirim et al., have studied a neural network scheme as a predictor for analyzing the transmission angle of a slider-crank mechanism with eccentric connector [14].

Erkaya and Uzmay have presented the optimization of the transmission angle for the slider-crank mechanism with joint clearances [15]. Tanik has introduced the transmission angle of a compliant slider-crank mechanism [16]. Similarity conditions for the transmission angle of the compliant slider-crank and its rigid body counterpart were derived via two theorems. Khorshidi et al., have presented a novel approach to the multi-objective optimal design of four-bar linkages for path-generation purposes [17]. Three, often conflicting criteria including the mechanisms tracking error, deviation of its transmission angle from 90° and its maximum angular velocity ratio are considered as objectives of the optimization problem.

A needle driving mechanism with optimized needle velocity and transmission force has a considerable influence on decreasing the fabric damages which may occur during stitch formation. To the best knowledge of the authors, no specific research has been carried out on this subject. Therefore, the main concern of this research is to present firstly a novel needle driving mechanism for which a precise optimization procedure has been performed. In addition, we will formulate the mechanical advantage of this compound mechanism for the first time through the location of the mechanism's instant centers. As the next contribution of this study, some of the most important design criteria such as reducing the needle generated heat, reducing undesirable vibrations and increasing the mechanical performance are considered by imposing different conflicting objective functions.

After acquiring an optimized mechanism with the demanded level of mechanical performance, further considerations may be carried out to adjust the parameters of other mechanisms of the sewing machine (hook, feed dog, and thread take-up lever mechanisms) to avoid any conflict in the operation of the machine in forming a proper stitch.

2 SLIDER CRANK MECHANISM

Needle movement in a standard lockstitch sewing machine is produced by a slider-crank mechanism. The geometry and structure of this mechanism is shown in Figure 1. The mechanism is defined through the drive link OA, the needle path s , the coupler link AB and the eccentricity e . Although this mechanism is very common due to its simple structure, it has some drawbacks with respect to the performance and kinematic behavior during sewing process [6]. Since design parameters of the slider crank mechanism are limited, one cannot implement major modifications to achieve high mechanical performance.

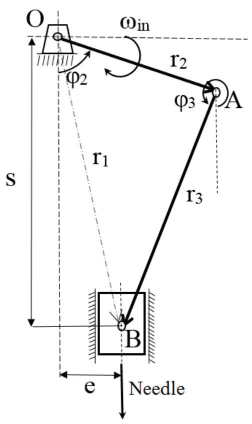

Fig. 1 The slider crank mechanism

3 KINEMATICS OF THE NEW MECHANISM

The newly developed mechanism has a more complex structure than the slider crank mechanism, as shown in Figure 2 [18]. This mechanism is the second generation of another complex mechanism which has been proposed recently by researchers to improve the mechanical performance of the sewing machine [6], [7]. Such mechanisms are typically composed of some elementary mechanisms including four or five links. Although our newly developed mechanism is different from the slider crank mechanism with respect to the kinematic parameters such as needle velocity, the stitch velocity would not be affected when compared to the slider crank sewing machines. The reason behind this claim is that, the process of one stitch in this mechanism is completed during one complete rotation of the input link of the sewing machine (360° rotation) similar to the slider crack mechanism. Therefore, the change of input link rotational velocity would result in the same change of stitch velocity in this mechanism. This means that the time needed for one complete stitch for the same input link rotational velocity in this mechanism and the slider crack mechanism is the same. On the other hand, the reduction of needle velocity in the penetration zone is compensated by increasing its value during the upward needle movement. In this way, no extra tension is exerted on the thread provided that necessary considerations are taken when designing the take-up lever mechanism. According to Figure 2, this mechanism consists of two mechanisms: a four-bar mechanism OABC and a five-bar slider crank mechanism OADE. The coupler link ABD is common in both mechanisms. The four-bar mechanism is included to restrict the total degrees-of-freedom (DOF) of the sewing machine to one. The crank OA is the input link and the slider E is the output link. For optimization purpose, it is mandatory to find the kinematic relations between the input crank OA and slider E (needle). Full description of the kinematic relations is presented in [18].

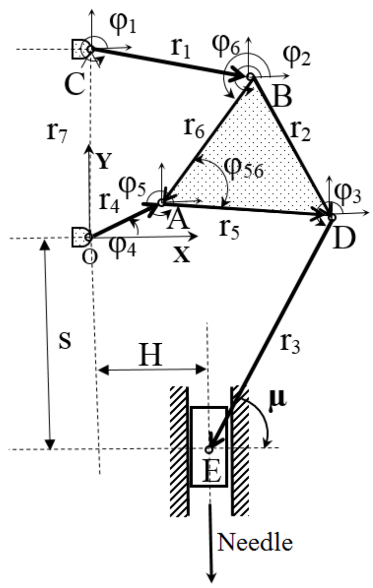

Fig. 2 The newly developed mechanism

4 TRANSMISSION ANGLE OF THE MECHANISM

Transmission angle (μ) is an important factor in mechanisms design. It denotes the quality of motion transmission in a mechanism and it is mainly used to obtain the better results for various linkage applications. A mechanism designed with an ideal transmission angle experiences minimum force acting along the coupler and on the bearings. Although a good transmission angle is not a cure-all for every design problem, it can guarantee improvement of the mechanism performance for many mechanical applications. When $\mu = 90^{\circ}$, the most effective force transmission takes place. Furthermore, the accuracy of output motion is less sensitive to manufacturing tolerances of link lengths and clearance between joints and change of dimensions due to thermal expansion. Mechanisms with transmission angle deviated from 90° exhibit poor operational characteristics like noise and jerk at high velocity [19], [20]. Transmission of motion is impossible when the transmission angle is 0° or 180°. In these cases, no load can be realized on output link. Thus the transmission angle of a mechanism provides a very good indication of the quality of motion, the accuracy of its performance, expected noise output and its costs in general. The recommended transmission angle is generally $90^\circ - 50^\circ \le \mu \le 90^\circ + 50^\circ$. The transmission angle does not consider the dynamic forces due to velocity and acceleration. So, for determination of the transmission characteristics of a linkage, it is not necessary to analyze forces and torque acting at each joint of the whole mechanism. According to Fig. 2, the transmission angle equation is defined as follows [19], [20].

$$\mu = \cos^{-1}\left(\frac{r_4\cos(\phi_4) + r_5\cos(\phi_5) - H}{r_3}\right) \tag{1}$$

5 MECHANICAL ADVANTAGE OF THE MECHANISM

One of the major aspects of which a designer must be aware is the ability for a particular mechanism to transmit torque or force. Some mechanisms, such as a gear train, transmit a constant torque ratio between the input and output because there is a constant velocity ratio between input and output. In a linkage, however, this is not the case. Two observations can be made without further analysis:

1. As hinted in the preceding mention of the gear train, the torque ratio is a function of the velocity or angular velocity ratio between output and input links of the mechanism.
2. The torque ratio is a function of geometric parameters, which, in the case of a linkage, will generally change during the course of the mechanism motion.

Since the angular velocity ratio can be expressed entirely in terms of directed distances (based on the instant-center approach), the mechanical advantage (MA) can be expressed completely in terms of ratios of distances. In many design situations, the mechanical advantage expression for a mechanism allows the optimal redesign of that device for performance improvement. In order to obtain a relation for the mechanical advantage, the instant center for each pair of two links of the mechanism is needed. Fig. 3 shows a number of instant centers of the mechanism.

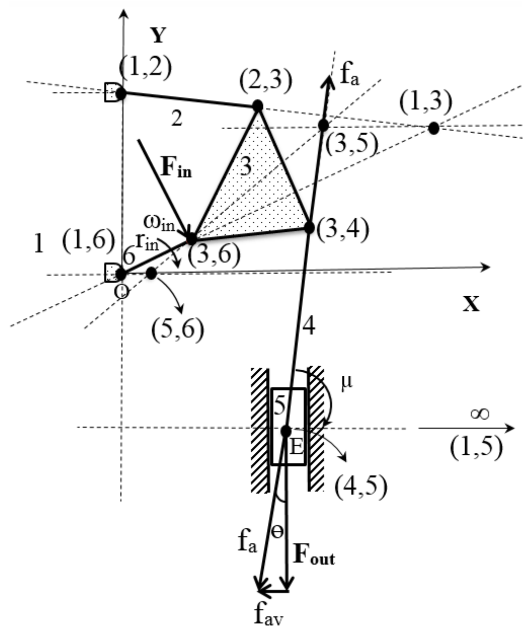

Fig. 3 The instant centers of the mechanism

Links 6 (OA) and 5 (slider E) are the input and output links, respectively. To obtain the mechanical advantage relation, we cannot use the presented method in mechanism design textbooks for the four bar mechanism (because link 5 is a slider with ω_5 = 0). Consequently, we begin to derive the mechanical advantage for our new mechanism. Assuming that the power input and power output are equal P_{in}=P_{out}, the following equation can be written

$$P_{in} = P_{out} \Rightarrow T_{in}\omega_{in} = F_{out}V_{out} \Rightarrow T_6\omega_6 = F_{out}V_E \tag{2}$$

Notice that since link 5 moves in the vertical slot, any point considered on it must have a velocity in the vertical direction. Moreover, all points of link 5 have the same velocity, including the point of the extended plane of link 5 which momentarily coincides with instant center (5,6). The velocity of this point can be written as:

$$V_E = \omega_6\left|\overrightarrow{(1,6-5,6)}\right| \tag{3}$$

By considering that $T_6 = r_{in}F_{in}$ and combining the above equations, the following relation is obtained.

$$(r_{in}F_{in})\omega_6 = F_{out}V_E$$

After substituting the slider velocity from Eq. (3) one obtains:

$$r_{in}F_{in}\omega_6 = F_{out}\omega_6\left|\overrightarrow{(1,6-5,6)}\right| \quad (4)$$

The mechanical advantage can finally be determined as:

$$MA = \frac{F_{out}}{F_{in}} = \frac{r_{in}}{\left|\overrightarrow{(1,6-5,6)}\right|} \quad (5)$$

6 GENETIC ALGORITHM

During the last decade, there has been a growing interest in obtaining the optimal solutions for complex systems using genetic algorithms (GA). Genetic algorithms are search algorithms that use the notions of natural selection and genetics [21], [22]. Genetic algorithms maintain a population of potential solutions and simulate evolution process using some selection process based on fitness of chromosomes and some genetic operators. The algorithm operates on population of results and with use of the principle of survival of the best and evolution, makes better solutions and suitable results (Fig. 4). Primarily, the initial population of specific chromosomes (strings) is generated randomly. Then, the objective function is used to evaluate chromosomes.

For the optimization problem, suitable chromosomes of the generation should provide the lowest value of the objective function. In order to let different chromosomes to be generated in specified conditions, combination and reproduction (crossover) steps are followed. Unlike the traditional method of the optimization, the genetic algorithm searches simultaneously in multiple points. The set of chromosomes which in each iteration of the algorithms are evaluated is called a generation. Details of the genetic algorithm used in this paper are shown as a flowchart in Figure 4.

7 THE OPTIMIZATION PROCESS

In this section, the optimization procedure of the newly developed mechanism based on the GA approach is presented. The objective here is to find the mechanism link lengths for which the design criteria are optimized.

A. Definition of the optimization problem

A good selection for the transmission angle in mechanism design is 90°, which is an ideal value. Minimizing deviation of the transmission angle from 90° results in an acceptable level of force transmission and mechanical advantage of the mechanism. Therefore, the objective function is sum of the squares of maximum and minimum deviations of transmission angle μ from 90°. The objective function is defined as follows [17]:

$$f_1 = \left[(\mu_{max} - 90^o)^2 + (\mu_{min} - 90^o)^2\right] \quad (6)$$

For introducing the objective function to the optimization procedure, following limitations for the mechanism geometry have to be considered:

- The Grashof's law has to be satisfied for mechanism OABC. According to this law, sum of the shortest and the longest link of a four-bar linkage has to be lower or equal to the sum of two remaining link lengths in order to be able to fully rotate the crank link.
- The distance between upper and lower dead points of the needle has to be constant within a small tolerance. This condition guaranties that the needle displacement remains constant regardless of the variation of link lengths.

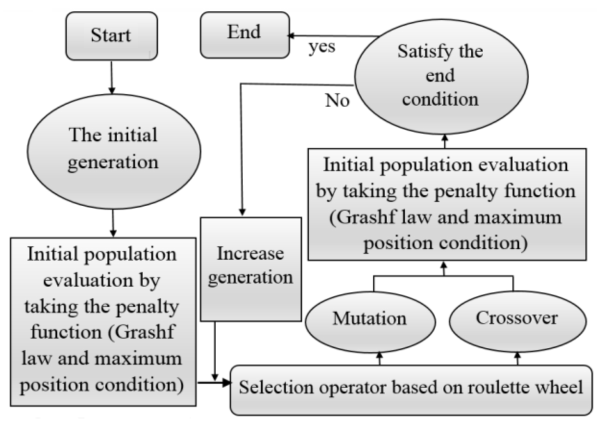

Fig. 4 Genetic Algorithm flowchart

In addition to these conditions, Table 1 summarizes some other optimization constraints regarding mechanism geometry. The optimization process is carried out in MATLAB 2011a. Table 2 shows the GA parameters used for initializing the optimization process.

Table 1 Optimization Conditions (all dimensions are in mm)

Parameter	Lower limit	Upper limit
r_1	13	18
r_2	43	50
r_3	40	45
r_4	7	12
r_5	38	42
r_6	37	43
r_7	36	45
H	10	15

B. Optimization results

The optimization of the mechanism was then performed to obtain the optimized link lengths which minimize the objective function, Eq. (6), by imposing the optimization conditions. Tables 3 and 4 show the optimal values of the mechanism link lengths, and the design criteria values, respectively.

Table 2 The GA Parameters

Number of iteration	100
Number of population	5000
Crossover percent	0.8
Mutation percent	0.3
Mutation rate	0.2
Objective function	Eq. (6)

Table 3 Optimal Link Lengths

Parameter	Optimal value (mm)
r_1	17.943
r_2	43.4
r_3	44.876
r_4	11.805
r_5	38.639
r_6	40.587
r_7	37.102
H	14.987

Table 4 Design Criteria Results

Design criteria	Value
Objective function value	419.74
MA in penetration zone	1.281
Needle Velocity in penetration zone $(\int_{\varphi_1}^{\varphi_2} \lvert\dot{s}(\varphi_4)\rvert d\varphi_4)$	344.77 (mm/s)
Penetration zone	$116^\circ \le \varphi_4 \le 145^\circ$

Figure 5 shows convergence of the genetic algorithm to achieve the optimal stable limit which may in several runs vary slightly according to the nature of the genetic algorithm. In this figure, value of the objective function is expressed according to the number of iterations (generations).

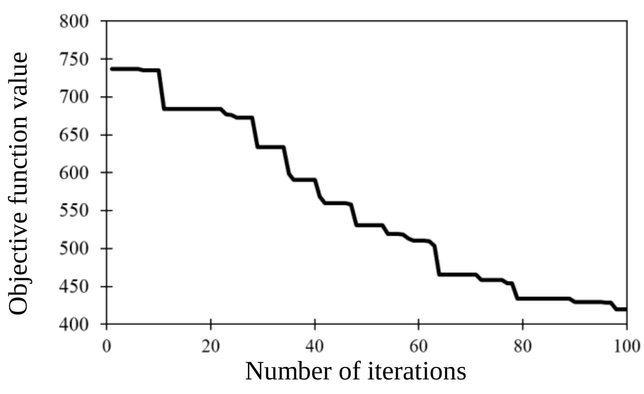

Fig. 5 Variation of the objective function value

The final value of the objective function is equal to 419.74. Depicted in Fig. 6 illustrates variation of the transmission angle during one rotation of the input link. It can clearly be seen that its value varies between 71° and 82.4° which are perfect for ideal operation of the mechanism.

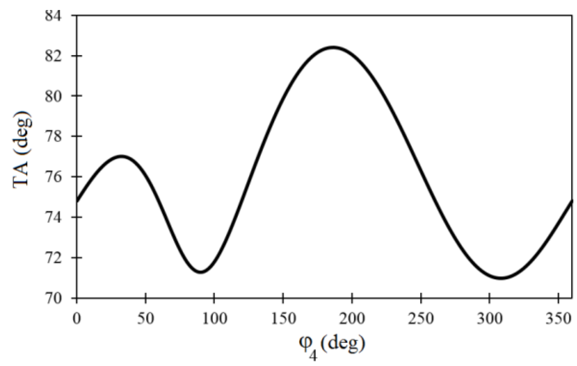

Fig. 6 Variation of the transmission angle

For our newly developed mechanism of the needle motion in [18], only sum of the needle velocities in the penetration zone $(\int_{\varphi_1}^{\varphi_2} \lvert\dot{s}(\varphi_4)\rvert d\varphi_4)$ has been introduced as the optimization objective. As mentioned previously, one of the effective factors to reduce the needle temperature is reduction of the needle velocity in the penetration zone [7]. Moreover, the needle velocity in the penetration zone affects the needle contact force in the fabric. On the other hand, optimizing the transmission angle has a considerable effect on the needle contact force. Therefore, the optimization process has been carried out here to optimize the transmission angle of the needle driving mechanism. The results of position (s), velocity ($\dot{s}$) and acceleration ($\ddot{s}$) of the needle for the optimized lengths in this study and optimization results in [18], are shown in Figs. 7, 8 and 9, respectively. The penetration zone in this study is associated with the range of $116^\circ \le \varphi_4 \le 145^\circ$. Furthermore, Table 5 shows the results of the optimized mechanism in [18] and the typical slider crank mechanism of Figure 1.

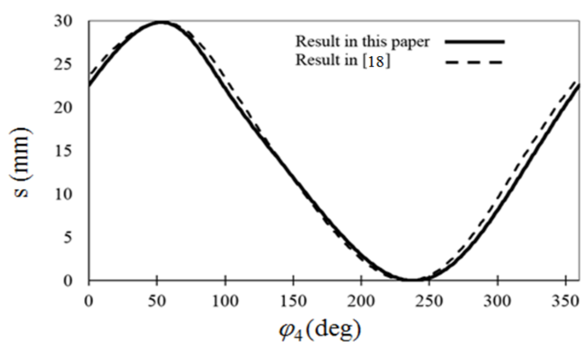

Fig. 7 Variation of the needle position

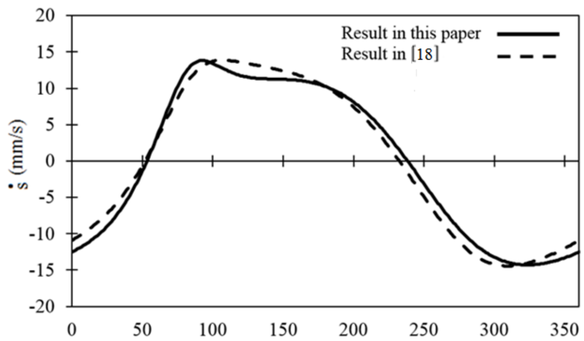

Fig. 8 Variation of the needle velocity

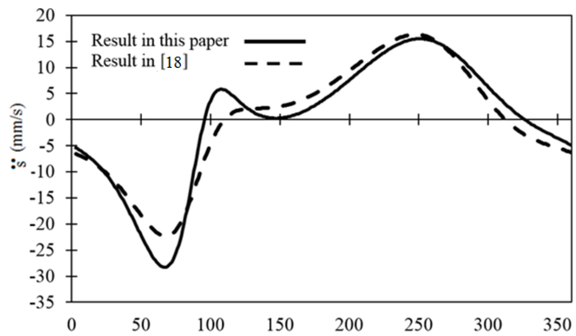

Fig. 9 Variation of the needle acceleration

Table 5 Design Criteria Results

	Result in [18]	The slider crank mechanism
$\int_{\varphi_i}^{\varphi_j} \lvert\dot{s}(\varphi_4)\rvert \mathrm{d}\varphi_4$	343.67	355.86
MA in the penetration zone	1.0821	1.0076

According to Figure 8, values of the needle velocity in the penetration zone for the optimized mechanism in this paper are less than their corresponding values of [18]. However, according to Tables 4 and 5, sum of the needle velocities in the penetration zone in this paper is slightly more than its value in [18]. The reason behind this increase is due to the fact that the optimized mechanism here experiences larger range of the penetration zone. On the other hand, the optimized mechanism here shows better performance with respect to decreasing the needle velocity when compared with the slider crank mechanism.
Based on Eq. (5), the mechanical advantage in the penetration zone is calculated for three different cases: (1) the optimized mechanism in this study, (2) the optimized mechanism of [18], and (3) for the slider crank mechanism (Tables 4 and 5). The results indicate the achievement of a higher mechanical advantage in this study compared to our previous work (15% higher than the optimized mechanism of [18] and 21% higher than the slider crank mechanism). Figure 10 shows variation details of the mechanical advantage which confirms the above mentioned statements.

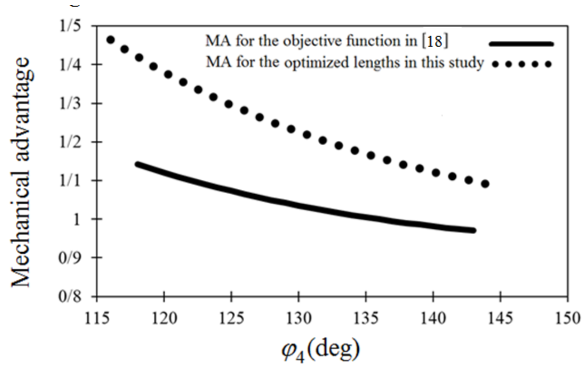

Fig. 10 Variation of the mechanical advantage in the penetration zone

8 THE MODIFIED OBJECTIVE FUNCTIONS

The optimization of the transmission angle (TA) based on the objective function defined in Eq. (6) resulted in a mechanism with higher mechanical advantage. However, it may cause some undesirable situations with respect to other design criteria. In order to remedy this drawback, we consider further the optimization problem by modifying the objective function which in addition to the optimization of transmission angle, some certain practical issues are considered. In this section, three other objective functions are introduced.

1- TA and sum of the absolute needle velocities in the penetration zone

Needle velocity in penetration zone directly affects the amount of needle heat generated during sewing process. Hence, reduction of the needle velocity in the penetration zone is very important. For this purpose, the objective function, Eq. (6), is modified to include minimization of this factor as follows:

$$f_2 = \mathrm{W}_1\left[(\mu_{\max} - 90^o)^2 + (\mu_{\min} - 90^o)^2\right] + \mathrm{W}_2\int_{\varphi_i}^{\varphi_j} \lvert\dot{s}(\varphi_4)\rvert d\varphi_4 \quad (7)$$

Weighting constants W_1 and W_2 should be chosen in such a way that both parts of the objective function share equal effects in obtaining the optimized mechanism.

2- TA and the mechanical advantage in penetration zone

A linkage with an excellent transmission angle in a particular configuration may not have a sufficient mechanical advantage. On the other hand, a linkage with a good mechanical advantage may have an unacceptable transmission angle. Since both the transmission angle and the mechanical advantage vary during linkage operation, either parameter can be critical to the designer. The objective function can be defined as follows:

$$f_3 = \mathrm{W}_1\left[(\mu_{\max} - 90^o)^2 + (\mu_{\min} - 90^o)^2\right] + \mathrm{W}_2 \frac{1}{\mathrm{MA}} \qquad (8)$$

3- TA, the needle velocity in penetration zone and the needle jerk ($\dddot{s}$)

According to Figure 9, it can be seen that improvement of the transmission angle results in higher variation of the needle acceleration which in turn may lead to undesirable vibrations of the sewing machine. Therefore, in addition to improvement of the transmission angle and reduction of the needle velocity in the penetration zone, reduction of the needle jerk during one rotation of the input link is greatly demanded. The objective function is consequently defined as follows:

$$\begin{aligned} f_4 = {} & \mathrm{W}_1\left[(\mu_{\max} - 90^o)^2 + (\mu_{\min} - 90^o)^2\right] + \\ & \mathrm{W}_2 \int_{\varphi_i}^{\varphi_j} |\dot{s}(\varphi_4)|\, d\varphi_4 + \mathrm{W}_3 \int_0^{2\pi} |\dddot{s}(\varphi_4)|\, d\varphi_4 \end{aligned} \qquad (9)$$

9 RESULTS

The optimization is performed here for the already mentioned three modified objective functions with the same conditions as stated before. The optimal link lengths of the mechanism are shown in Table 6. The results of the optimization are presented in Table 7. Comparing different values of the needle velocity reveals that considering these objective functions results in almost the same values. However, its corresponding value of the objective function f_3 is slightly higher as it does not include any needle velocity.

On the other hand, imposing different objective functions shows considerable effect on the needle jerk. The objective function f_4 leads to a 29.4% reduction of the needle jerk with respect to f_3. The interesting point which may be realized by comparing the terms included in f_2 and f_3 is that, improving the mechanical advantage of about 14% causes an increase in the value of the needle jerk about 30%. This clearly states that achieving a higher mechanical advantage is gained by the cost of increasing the needle jerk which in turn yields undesirable vibrations. Finally, it is clearly observed from Table 7 that objective function f_3 shows better performance with respect to the improvement of the mechanical advantage.

Table 6 Optimal Link Lengths with Modified Objective Functions (all dimensions are in mm)

Parameter	Optimal value in f_2	Optimal value in f_3	Optimal value in f_4
r_1	17.966	17.905	17.935
r_2	45.625	43.351	45.701
r_3	44.174	45	41.57
r_4	11.768	12	11.354
r_5	40.597	38.446	41.529
r_6	43	40.99	41.69
r_7	39.374	37.169	40.094
H	12.112	14.986	13.154

Table 7 Design Criteria Results for the Objective Functions

Design criteria	Objective function f_2	Objective function f_3	Objective function f_4
Needle velocity in the penetration zone ($\int_{\varphi_i}^{\varphi_j} \lvert\dot{s}(\varphi_4)\rvert d\varphi_4$ (mm/s))	343.7287	344.7541	343.9719
Needle jerk ($\int_0^{2\pi} \lvert\dddot{s}(\varphi_4)\rvert d\varphi_4$ (mm/s³))	91833.9716	109576.2605	77303.3374
MA in the penetration zone	1.1428	1.2875	1.1115
Penetration zone	$117^\circ \le \varphi_4 \le 144^\circ$	$115^\circ \le \varphi_4 \le 145^\circ$	$123^\circ \le \varphi_4 \le 148^\circ$

Figs. 11 and 12 illustrate variation of the needle jerk and mechanical advantage, respectively. As it can be observed, minimizing the needle velocity in [18], and objective function f_3 represent respectively the best and the worst performance with respect to the reduction of the needle jerk. On the other hand, the mechanical advantage obtained by considering f_3 shows a clear superiority compared to other objective functions. This situation clearly states that when obtaining a high level of mechanical advantage is required, the objective function f_3 is preferred. However, in cases that in addition to achieving a high mechanical advantage, reducing the needle jerk is of great importance, one can select f_2 as the objective function.

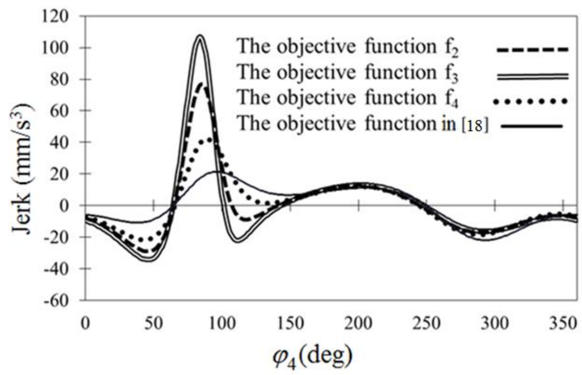

Fig. 11 Variation of the needle jerk

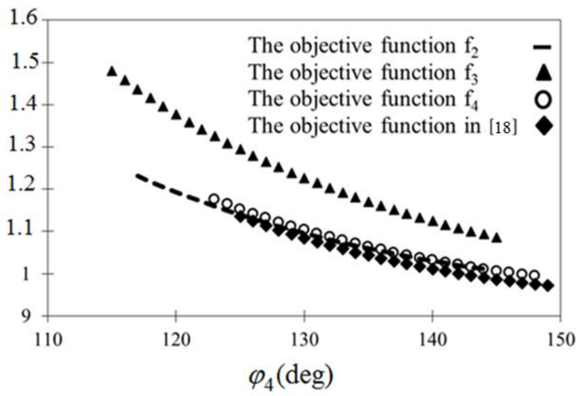

Fig. 12 Variation of the mechanical advantage in the penetration zone

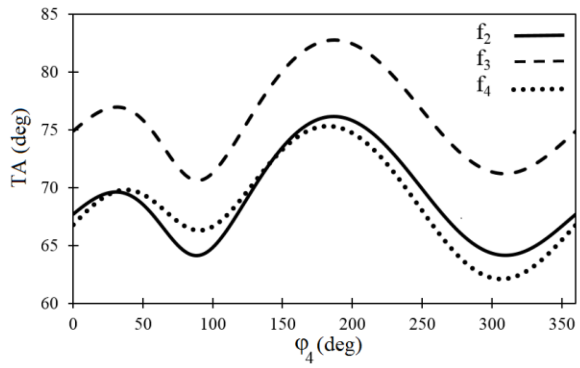

Fig. 13 Variation of the transmission angle

The above outlined results can further be confirmed by comparing variation of the transmission angle for three modified objective functions. According to Figure 13, the transmission angle in the mechanism which is optimized based on f_3 is closer to 90°. As stated before, minimizing deviation of the transmission angle from 90° results in acceptable level of force transmission and mechanical advantage of the mechanism. Optimizing both the transmission angle and the mechanical advantage in f_3 provides this demanded situation.

10 CONCLUSION

In this paper, after introducing a newly developed needle driving mechanism of a sewing machine, optimization of some design criteria using genetic algorithm was presented. For this purpose, first the notions of transmission angle and mechanical advantage of a mechanism were presented. Following an approach based on the location of the mechanism's instant-centers, derivation of the mechanical advantage relation was carried out. In the next step, the multiobjective constrained optimization process of this mechanism using the genetic algorithm was explained. It was observed that imposing different objective functions has considerable effect on the needle jerk and the mechanical advantage.

As an interesting point, it was concluded that maximizing the mechanical advantage causes a remarkable increase in the value of needle jerk. This clearly states that achieving a higher mechanical advantage is gained by the cost of increasing the needle jerk. Results confirm improvement of the required design criteria of the newly developed mechanism in this study. Based on these presented results, our newly developed mechanism operates much better than the typical slider-crank mechanism which is widely used in the sewing machines.

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