Geometric Optimization of the Five-Point Double-Toggle System for the Clamping Unit of an Injection Molding Machine with Response Surface Methods

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ABSTRACT

Injection molding is one of the most used processes for the manufacture of plastic products with high-volume capacity. The five-point double toggle mechanism is frequently employed in the clamping unit to augment the force generated by a linear actuator, thereby achieving the requisite clamping force. This is a crucial aspect that determines the productivity and quality of the finished product in the injection molding process. Nevertheless, the geometric synthesis of this type of mechanism has not been sufficiently addressed with regard to its integration into the mold design and the opening/closing stroke. This paper presents a novel approach for analyzing and evaluating the impact of parameters pertaining to mechanism posture on its force amplification, employing the Taguchi method. The relationship between force amplification ratio and the most influential parameters is simplified with the Face-Centered Central Composite Design (FCCCD) method, thereby allowing the optimal posture of the mechanism at the mold-closing stage to be determined. With these optimal parameters, once the mold height and desired opening/closing stroke have been selected, the dimensions of the links in the mechanism can be calculated. The results demonstrate that there are numerous combinations of these parameters that can yield a high force amplification ratio, thus providing the designer with a range of options for the design of a clamping unit. The proposed method can be employed at the initial stage of the design process to obtain a preliminary design, thus preparing for the dynamic analysis or further optimization problems of the mechanism.

Keywords-injection molding; clamping mechanism; Taguchi method; five-point; double toggle mechanism; Face-Centered Central Composite Design (FCCCD)

I. INTRODUCTION

An injection molding machine is a sophisticated apparatus comprising a multitude of intricate modules. Two essential components of an injection molding machine are the plasticizing unit and the clamping unit. The components of each unit, accompanied by their respective identification numbers, are shown in Figure 1. The clamping unit is a crucial component of an injection molding machine and is used to maintain the spatial relationship between the two halves of the mold, generate the requisite clamping force during the injection of the plastic into the mold, and facilitate the opening of the mold for the ejection of the molded object once the cooling process is complete. A variety of mechanisms are employed in the construction of a clamping unit [1]. Among these, the five-point double-toggle mechanism, also known as the nine-link double-toggle mechanism, is the most prevalent in modern injection molding machines. Researchers have examined the dynamic behavior of this type of mechanism and attempted to optimize its design with the objective of enhancing its efficiency, reducing the vibration of the system, and minimizing the input force provided by the linear actuator. One challenge inherent to the design phase is the kinematics and dynamics analysis of this mechanism. The result of this analysis is frequently a set of equations that describe the relationship between the input motion/force provided by a hydraulic cylinder and the output motion/force of a movable platen. An understanding of the dynamic behavior of the mechanism is fundamental to the optimization of its geometry, involving link dimensions, shape, etc., and the design of a controller for the clamping unit. Authors in [2] conducted a kinematic and dynamic analysis of the four-point and five-point toggle mechanisms, employing Hamilton's principle and the Lagrange multiplier method. The simulation demonstrated that the four-point type mechanism was capable of attaining the desired output force with a smaller input force than the five-point mechanism. A study on the effect of various types of input motion on the resulting output motion of a movable platen is presented in [3].

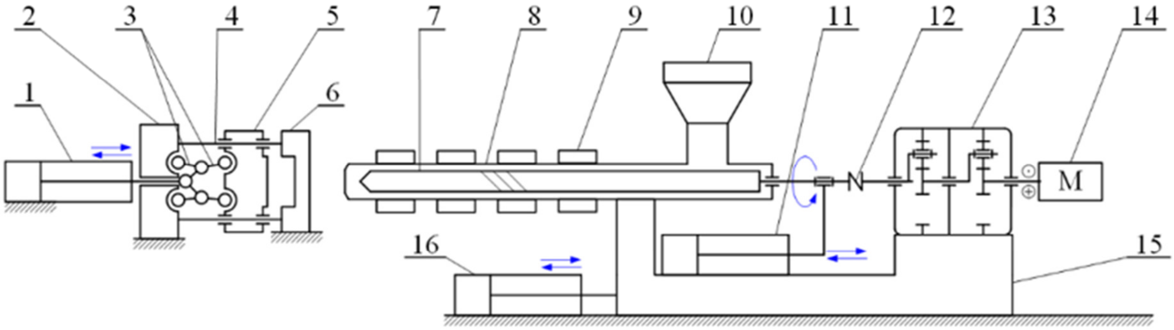

Fig. 1. Diagram of injection molding machine: hydraulic cylinder (1,11,16), rear platen (2), toggle system (3), tie bar (4), movable platen (5), stationary platen (6), plasticizing screw (7), barrel (8), band heater (9), hopper (10), coupling (12), planetary gear reducer (13), electrical motor (14), sliding support base (15).

In that study, the authors examined the kinematics of a clamping mechanism through two distinct analytical approaches: a constant speed analysis and a variable speed adjustment of a cross-head connected to a cylinder rod. The study indicated that while the constant speed approach could facilitate the mold-closing stage, it could also compromise the stability of the mold when the cross-head speed was high. Conversely, the variable speed approach may prove more intricate than the constant speed one, given the necessity of incorporating multiple acceleration and deceleration phases. By making reasonable adjustments to the ending position at the acceleration stage and the starting position of the deceleration stage, it is possible to control the stability of the mold. The influence of lubrication clearance on the five-point mechanism's revolute joints was examined in [4]. A dynamic analysis of the mechanism with lubricated clearance was conducted and compared with that of dry clearance. The authors concluded that the application of an appropriate oil viscosity through lubrication could serve to reduce the vibration and increase the stability of the system. Authors in [5], proposed a method for the analysis of mechanical error in a toggle mechanism in an injection molding machine, which assists engineers in the design and selection of appropriate link tolerances, thereby enabling the attainment of the desired motion accuracy of a clamping unit. Authors in [6], presented a simulation of the dynamics of a five-point toggle mechanism, taking into account the effects of inertia forces of links, friction at joints, and other factors at different stages of the clamping process. The simulation results and measurements obtained from the actual machine operation exhibited a high degree of correlation. The optimization of mechanism dimensions to achieve higher efficiency has also been a subject of research. Authors in [7, 8] conducted a parametric study to identify optimal designs for both conventional and Fanuc's five-point double-toggle mechanism. Additionally, they proposed a hybrid algorithm, designated as the GA-DE, which combined a real-value genetic algorithm with differential evolution. This approach was proposed as an alternative to parametric studies in the optimization of mechanism design, with the objective of reducing the thrust required to apply to the crosshead [9]. Authors in [10], demonstrated the optimization of mechanism parameters, including the speed profile of the moving platen, the stroke of the hydraulic cylinder in comparison to the mold-opening stroke, and the ratio of Force Amplification (FA) and the initial angle of mold closing, employing the genetic algorithm. The dynamic analysis of the mechanism included an examination of the friction forces acting on the sliders and the friction moments present in the revolute joints. A comprehensive investigation was conducted to ascertain the impact of link dimensions and angle on the speed profile, mold-opening stroke, and the requisite force from a hydraulic cylinder. The optimized result yielded a higher force amplification ratio, a higher stroke ratio, and a smoother speed profile. Authors in [11, 12] presented several variations of the toggle mechanism for the clamping unit, together with a kinematic analysis conducted using the vector loop method. Additionally, authors in [13], conducted a study on the optimal design of the Watt-chain double-toggle mechanism (five-point double toggle mechanism) with the objective of reducing the maximum acceleration and increasing the force amplification ratio. Authors in [14], employed the software MotionView for dynamic simulation and HyperStudy for digital graph analysis in an attempt to optimize the geometry of a mechanism.

The dynamic analysis and optimization of the five-point double-toggle mechanism requires a significant investment of time and effort to establish and solve a complex system of equations. This can be achieved when sufficient information is provided, including geometric dimensions, masses, and moment of inertia of links, as well as friction coefficients between mating components. However, this information is typically unavailable at the initial stage of the design process. Accordingly, the objective of this study is to present a more straightforward methodology for the design of a five-point double-toggle mechanism. The position equations of the mechanism are derived in order to describe the relationship between link dimensions and the posture of the mechanism. The relationship between the input force provided by the linear actuator (hydraulic cylinder) and the force exerted on the movable platen in the injection stage - opposite to the clamping force - is obtained through a force analysis with vector statics. This relationship can be expressed as a force amplification ratio. It is determined that, with the same requisite clamping force, an elevated force amplification ratio will result in a reduction of the necessary input force. The implementation of a mechanism with a high force amplification ratio has the potential to reduce the size of the hydraulic cylinder, thereby reducing the overall costs associated with the injection molding system. Accordingly, this study is concerned with the optimization of the force amplification ratio for the toggle mechanism. At the closing stage, the force acting on the movable platen is significantly greater than the friction forces in joints and inertia forces of the links, therefore these forces are not included in the analysis. The Design of Experiment (DOE) method has demonstrated its efficacy in optimization problems with multiple variables with a reduced expenditure of effort in comparison to the aforementioned methods [15]. In this study, the Taguchi method is deployed to ascertain the influence of the geometric parameters of the mechanism on the force amplification ratio at the closing stage, with the objective of identifying the factors that affect this ratio (screening design). Furthermore, the FCCCD response surface is employed to facilitate the correlation between the force amplification ratio and the predominant geometric parameters, transforming a nonlinear function into a second-order algebraic function. By specifying the dimensions of the movable platen, the opening and closing strokes, and the initial and final positions of the cross-head of the cylinder or movable platen, the designer can calculate the dimensions of the links in the mechanism.

II. THEORETICAL BACKGROUND

The kinematic diagram of the mechanism is presented in Figure 2. Given that the mechanism comprises two symmetric groups of links - the upper group includes slider A, link AB, link BCD, link DE, and slider E, the lower group entails slider A, link AI, link IHG, link GF, slider F, and the ground is a machine frame - an initial analysis of the kinematics of the upper group is warranted. In the absence of consideration of the impact of dimensional inconsistencies in the links on the mechanism's kinematics [5], the links in the lower group exhibit certain kinematic attributes analogous to those observed in their counterparts in the upper group. For rotational links and their counterparts, such as the pair of links AB and AI, the links BCD and IHG, and the links DE and GF, the rotational speeds are identical but the direction of rotation is opposite. The velocities of sliders A and E are identical. The upper group is then separated, and the coordinate system is set in accordance with the specifications presented in Figure 3. The position equations of the mechanism are:

$$\begin{cases} BC\cos(\varphi_3+\alpha) - AB\cos\varphi_2 = x_c \\ BC\sin(\varphi_3+\alpha) + AB\sin\varphi_2 = H \end{cases} \tag{1}$$

$$\begin{cases} CD\cos\varphi_3 + DE\cos\varphi_4 = x_m \\ CD\sin\varphi_3 - DE\sin\varphi_4 = e \end{cases} \tag{2}$$

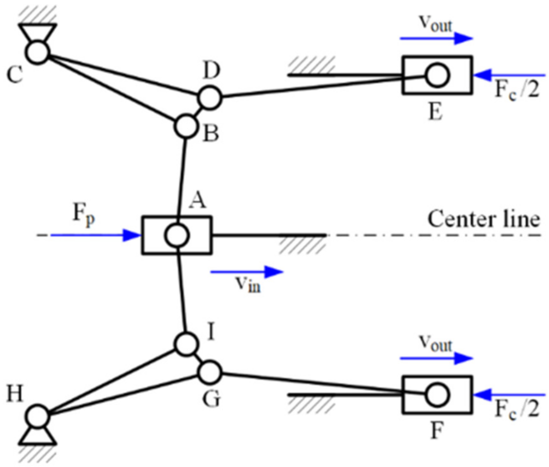

Fig. 2. Kinematic diagram of five-point double-toggle mechanism.

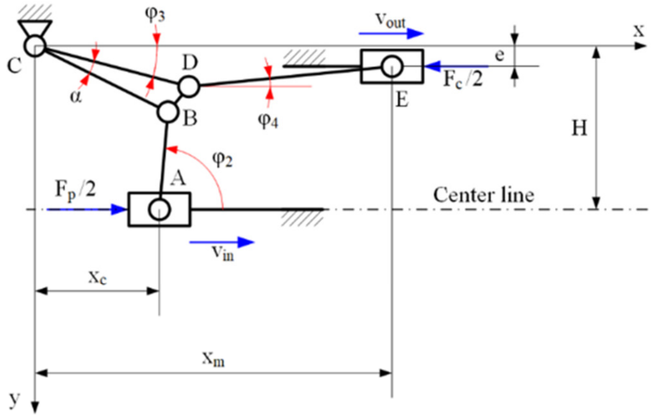

Fig. 3. Kinematic diagram of upper half of mechanism.

In order for the movable platen to travel with a stroke length of L (mm) within a time interval of T (s), the condition for the posture of the mechanism is:

$$\begin{cases} t_1 = 0: x_m(t_1) = a \\ t_2 = T: x_m(t_2) = a + L \end{cases} \tag{3}$$

where a (mm) is the initial position of slider E and also the smallest distance between the movable and the rear platen.

In the preliminary stages of the design process, the mass and moment of inertia of each link are undefined, as the shape and dimensions of the link have yet to be determined. In order to facilitate the analysis, the force applied to each link is considered in isolation, without consideration of the effects of gravitational, inertial, and frictional forces. The input force exerted by the cross-head of the cylinder is symbolized as F_p and acts on slider A. This force is transmitted and amplified by the mechanism to create the clamping force acting on sliders E and F (movable platen) to maintain the mold's closure during the injection of plastic at high pressure. The net force resulting from the pressure applied to the mold cavity is in equilibrium with the clamping force generated by the mechanism F_c. To analyze the forces on the mechanism, the method of vector statics [16] is employed. This entails releasing the constraint between two links and replacing it with reaction forces acting on each link. This results in three equations of force equilibrium for each link, which are then solved. In the case of a revolute joint, the reaction forces are two forces whose directions are parallel to the C_x and C_y axes, respectively and the notation R_{ab}^x represents the reaction force with a direction parallel to the C_x axis, exerted on link b by link a. Similarly, R_{ab}^y denotes the reaction force with a direction parallel to the C_y axis, exerted on link b by link a. In the case of a prismatic joint, the reaction forces are considered to be one force with a direction normal to the translational direction and one moment acting on the slider. The reaction forces are represented by N_{ab}, which denotes a reaction force with a direction normal to the translational direction, exerted on link b by link a. Similarly, M_{ab} represents a reaction moment exerted on link b by link a. It should be noted that the directions of the reaction forces are unknown; therefore, their directions are assumed to be as shown in Figure 4. Upon solving the system of equations, it was determined that the forces and moments with negative values indicate that their correct directions are opposite to the assumed directions. The free body diagram and the force balance equations for each link are:

Slider A (link 1):

$$\begin{cases} \sum F_x = F_p/2 - R_{21}^x = 0 \\ \sum F_y = -N_{01} + R_{21}^y = 0 \\ \sum M_z = M_{01} = 0 \end{cases} \tag{4}$$

Link AB (link 2):

$$\begin{cases} \sum F_x = R_{12}^x - R_{32}^x = 0 \\ \sum F_y = R_{12}^y - R_{32}^y = 0 \\ \sum M_{z/A} = -AB \sin\varphi_2 R_{32}^x + AB \cos\varphi_2 R_{32}^y = 0 \end{cases} \tag{5}$$

Link BCD (link 3):

$$\begin{cases} \sum F_x = R_{03}^x + R_{23}^x - R_{43}^x = 0 \\ \sum F_y = -R_{03}^y + R_{23}^y - R_{43}^y = 0 \\ \sum M_{z/C} = -CD \sin\varphi_3 R_{43}^x - CD \cos\varphi_3 R_{43}^y \\ +BC \sin(\varphi_3 + \alpha) R_{23}^x + BC \cos(\varphi_3 + \alpha) R_{23}^y = 0 \end{cases} \tag{6}$$

Link DE (link 4):

$$\begin{cases} \sum F_x = R_{34}^x - R_{54}^x = 0 \\ \sum F_y = R_{34}^y - R_{54}^y = 0 \\ \sum M_{z/D} = DE \sin\varphi_4 R_{54}^x - DE \cos\varphi_4 R_{54}^y = 0 \end{cases} \tag{7}$$

Slider E (link 5):

$$\begin{cases} \sum F_x = R_{45}^x - F_c/2 = 0 \\ \sum F_y = R_{45}^y - N_{05} = 0 \\ \sum M_{z/E} = M_{05} = 0 \end{cases} \tag{8}$$

Upon solving (4)-(8), the relationship between the clamping force and the input force of the hydraulic cylinder is obtained:

$$FA = \frac{F_c}{F_p} = \frac{BC}{CD} \frac{\sin(\varphi_3 + \alpha) + \tan\varphi_2 \cos(\varphi_3 + \alpha)}{\sin\varphi_3 + \tan\varphi_4 \cos\varphi_3} \\ = f(\varphi_2, \varphi_3, \varphi_4, \alpha, BC/CD) \tag{9}$$

where FA is the force amplification ratio. The constraints for φ_2, φ_3, φ_4, α, and BC/CD at the mold-closing position are given by the statistics in Table I.

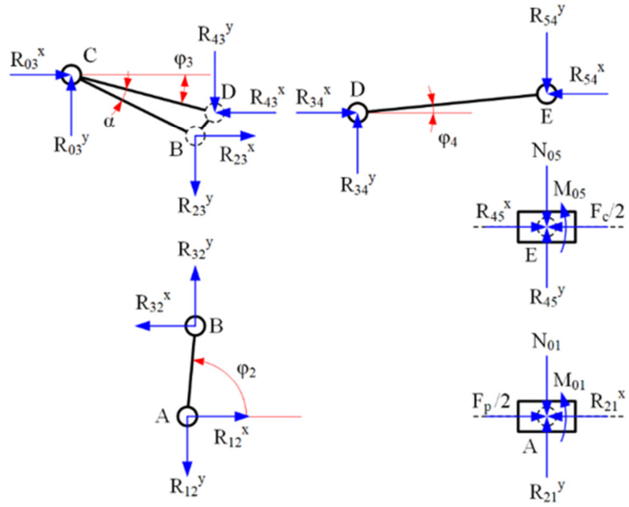

Fig. 4. Force analysis on each link.

TABLE I. LOWER AND UPPER LIMIT OF FACTORS

Factor	Chosen		From reference			Reference
	Lower limit	Upper limit	Lower limit	Upper limit	Specified value	
φ_2	83°	89°	70°	90°	-	[10]
			-	-	74.82°	[14]
φ_3	2°	5°	0°	4°	-	[8]
			-	-	4°	[14]
φ_4	2°	5°	-	-	3.07°	[14]
α	0°	30°	0°	28.49°	-	[9]
			10°	30°	-	[10]
BC/CD	0.4	0.8	0.3	1	-	[9]
			0.5	0.9	-	[10]

III. CALCULATION AND RESULTS

Once the relationship between the force amplification ratio and the geometric parameters of the mechanism was established, the Taguchi method was deployed to determine the most influential factor(s) (screening design). Subsequently, the response surface method, FCCCD, was used to ascertain the dependence of FA on these factors. Finally, the optimal combination of factors was identified in order to achieve the desired FA. Initially, the Taguchi L16 with five factors, φ_2, φ_3, φ_4, α, *BD/CD*, and four levels was employed to assess the impact of each factor on the ratio in question. The denotations and levels of the factors used in the Taguchi analysis are presented in Table II.

TABLE II. LEVELS OF FACTORS FOR TAGUCHI ANALYSIS

Factor	Value level according to Taguchi				Difference
	1 (Min_	2	3	4 (Max)	
X_1 *(angle φ_2)*	83°	85°	87°	89°	6°
X_2 *(angle φ_3)*	2°	3°	4°	5°	3°
X_3 *(angle φ_4)*	2°	3°	4°	5°	3°
X_4 *(angle α)*	0°	10°	20°	30°	30°
X_5 *(ratio BC/CD)*	0.4	0.53	0.67	0.8	0.4

In order to calculate the values of FA, (9) was used with the factor values in the design matrix of the Taguchi method L16 and the results are portrayed in Table III. The Minitab analysis of the Taguchi design of FA versus x_1, x_2, x_3, x_4 and x_5, yielded the results depicted in Table IV and Figure 5.

TABLE III. DESIGN OF EXPERIMENT MATRIX FOR TAGUCHI METHOD

No.	x_1	x_2	x_3	x_4	x_5	*FA*
1	1	1	1	1	1	46.845
2	1	2	2	2	2	41.580
3	1	3	3	3	3	37.497
4	1	4	4	4	4	33.251
5	2	1	2	3	4	100.577
6	2	2	1	4	3	77.444
7	2	3	4	1	2	38.963
8	2	4	3	2	1	28.822
9	3	1	3	4	2	85.060
10	3	2	4	3	1	51.408
11	3	3	1	2	4	143.462
12	3	4	2	1	3	91.347
13	4	1	4	2	3	306.514
14	4	2	3	1	4	374.987
15	4	3	2	4	1	157.509
16	4	4	1	3	2	228.936

TABLE IV. ANOVA ANALYSIS RESULTS

Level	x_1	x_2	x_3	x_4	x_5
1	31.93	40.45	40.38	38.98	35.19
2	34.71	38.96	38.90	38.61	37.49
3	38.79	37.59	37.69	38.24	39.55
4	48.09	36.51	36.55	37.69	41.28
Delta	16.16	3.94	3.83	1.29	6.08
%	51.63	12.59	12.23	4.12	19.43
Rank	1	3	4	5	2

The results of the ANOVA analysis indicate that factors x_1 and x_5 have a notable impact on force amplification, with respective contributions of 51.63% and 19.43%. The remaining factors, x_2, x_3, and x_4, exhibit considerably less pronounced effects, estimated at 12.59%, 12.23%, and 4.12%, respectively. The impact of each factor on *FA* is also shown in Figure 5. The optimal *FA*, as determined by the Taguchi method, can be achieved when the coded variables are set to $x_1 = 4$, $x_2 = 1$, $x_3 = 1$, $x_4 = 1$, and $x_5 = 4$. Alternatively, for uncoded variables, the optimal settings are $\varphi_2 = 89°$, $\varphi_3 = 2°$, $\varphi_4 = 2°$, $\alpha = 0°$, and $BC/CD = 0.8$. Based on these findings, the FCCCD was applied to develop a regression model of FA, with φ_2 and $u = BC/CD$ as the two factors. This enabled the determination of the value ranges for both factors that yielded the highest or most suitable FA. In this case, the values of the other factors were fixed, including $\varphi_3 = 2°$, $\varphi_4 = 2°$, and $\alpha = 0°$. As evidenced in Figure 5, there is a notable increase in *FA* when φ_2 is within the range of 83° to 89° and *BC/CD* is between 0.4 and 0.8. Accordingly, these ranges are designated as the lower and upper limits for the selected factors in FCCCD as presented in Table V. With the values of the factors φ_2 and u in the design matrix of FCCCD and the fixed values of φ_3=2°, φ_4=2°, and α=0°, the *FA* values were calculated, and the results are illustrated in Table VI.

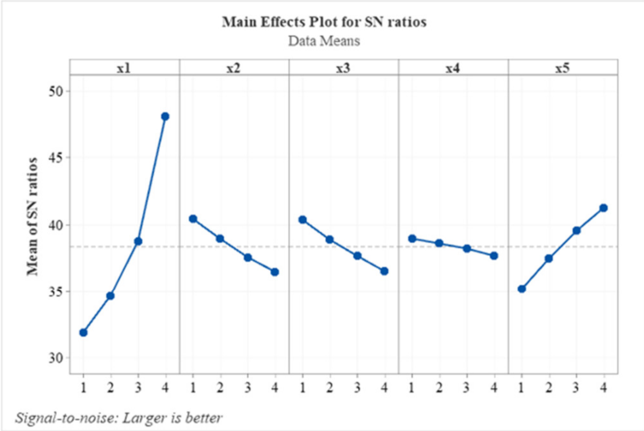

Fig. 5. Main effects plot for SN ratios.

TABLE V. FACTOR LEVELS FOR FCCCD METHOD

Coded Factor	Uncoded Factor	Low	Middle	High
		-1	0	+1
x_1	φ_2	83°	86°	89°
x_2	$u = BC/CD$	0.4	0.6	0.8

TABLE VI. DESIGN MATRIX FOR FCCCD

No.	Coded		Uncoded		FA
	x_1	x_2	φ_2	u	
1	-1	-1	83°	0.4	46.845
2	+1	-1	89°	0.4	328.314
3	-1	+1	83°	0.8	93.689
4	+1	+1	89°	0.8	656.628
5	-1	0	83°	0.6	70.267
6	+1	0	89°	0.6	492.471
7	0	-1	86°	0.4	82.104
8	0	+1	86°	0.8	164.207
9	0	0	86°	0.6	123.155

We used Minitab to obtain the regression model of *FA* as a second-order algebraic function:

$$FA = 129911 - 3024\varphi_2 - 9705u + 17.58\varphi_2^2 + 117.3\varphi_2 u \quad (10)$$

As indicated in the model summary provided by Minitab, the standard deviation of the model is 35.1586. The *R*-squared value of the regression model is 99.01%. This suggests that the regression model is an appropriate fit for the specified data set. Figure 6 presents the surface plot of *FA*, which provides a visual representation of the variation in *FA* across the ranges of φ_2 and *BC/CD*. As φ_2 and *BC/CD* increase, *FA* also increases. The contour plot, based on the regression model of *FA*, as observed in Figure 7, indicates the range of φ_2 and *BC/CD* that can yield the same force amplification. This allows the designer to consider a variety of options for a five-point double-toggle mechanism with a specified force amplification ratio. From the regression model, it can be predicted that the maximum value of *FA* will be 639.0486 at the boundary of φ_2=89° and u=0.8. Once the geometric parameters of the system are established, the dimensions of the links in the mechanism can be calculated. The mold height (2*H*), opening/closing stroke (*L*), distance between the movable and rear platens (a), and eccentricity (e) can be specified by the user or selected by the designer.

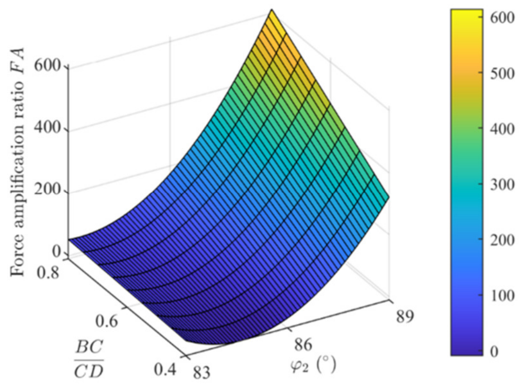

Fig. 6. Surface plot of *FA*.

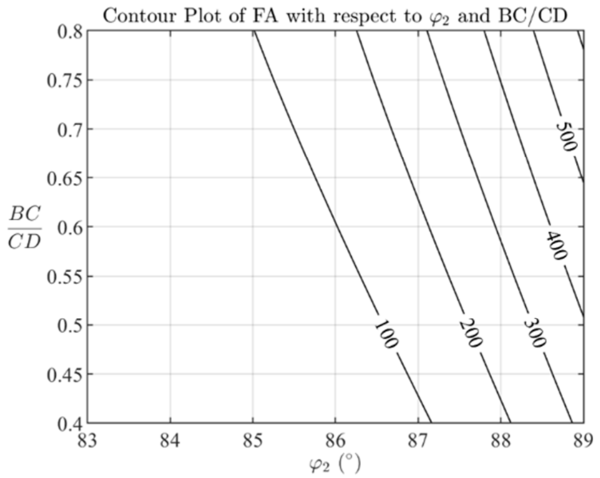

Fig. 7. Coutour plot of *FA*.

The dimensions of the mold clamping area were standardized in [17]. Examples of these dimensions include the following values: 200 mm, 224 mm, 250 mm, 280 mm. By employing the systems of (1) and (2) in conjunction with the ratio $u=BC/CD$, it is possible to calculate the dimensions of the links:

$$\begin{bmatrix} AB \\ BC \\ CD \\ DE \end{bmatrix} = \begin{bmatrix} \sin\varphi_2 & \sin(\varphi_3+\alpha) & 0 & 0 \\ 0 & 1 & -u & 0 \\ 0 & 0 & \cos\varphi_3 & \cos\varphi_4 \\ 0 & 0 & \sin\varphi_3 & -\sin\varphi_4 \end{bmatrix}^{-1} \begin{bmatrix} H \\ 0 \\ x_m \\ e \end{bmatrix} \quad (11)$$

The selected mold height (H) was 280 mm, the distance between the movable and rear platen at the mold opening position (a) was set at 300 mm, the stroke (L) was selected to be 300 mm, the eccentricity (e) was set to zero, the angles of link CD and link DE with respect to horizontal line (φ_3 and φ_4) were set to 2° each, and the angle (a) was set to zero. Based on these parameters, the dimensions of the links were calculated and the kinematic diagram of the mechanism was drawn for two cases:

- In Case 1, with φ_2=83°, u=0.6, and FA=70.267, the dimensions of the links are calculated as: AB=275.770 (mm), BC=180.110 (mm), CD=DE=300.183 (mm). The dimensions are rounded to the nearest tenth: AB=275.8 (mm), BC=180.1 (mm), CD=DE=300.2 (mm). Subsequently, a diagram of the mechanism was constructed, as shown in Figure 8, and the value of FA was recalculated at 62.514.
- In Case 2, with φ_2=86°, u=0.6, and FA=123.1, the dimensions of the links are calculated as: AB=274.382 mm, BC=180.110 mm, CD=DE=300.183 mm. The aforementioned dimensions are rounded to the nearest tenth: AB=274.4 (mm), BC=180.1 (mm), CD=DE= 300.2 (mm). Subsequently, the diagram of the mechanism, presented in Figure 9, was constructed, and a further recalculation of FA was performed that was finally adjusted at 97.582.

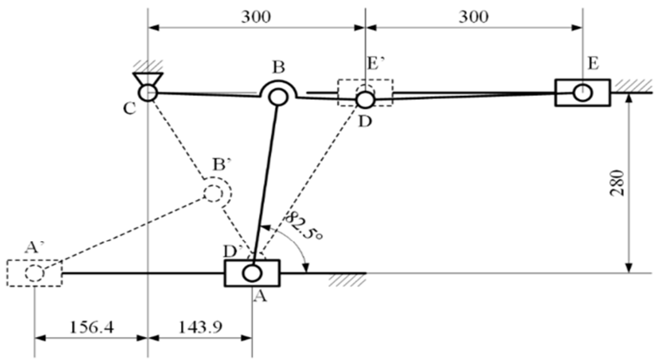

Fig. 8. Kinematic diagram of mechanism (case 1).

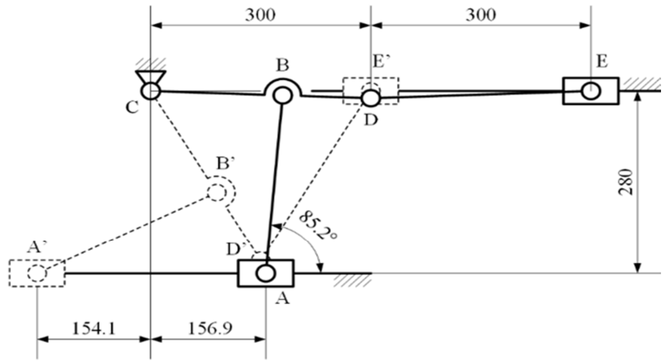

Fig. 9. Kinematic diagram of mechanism (case 2).

IV. DISCUSSION

In the design of a five-point double-toggle mechanism, a variety of methods have been developed to identify the optimal geometric profile of the mechanism. These include parametric design and genetic algorithms, among others. Nevertheless, these methods necessitate a considerable investment of effort and time, such as parametric design [7,8], or sophisticated algorithms, such as genetic algorithms [9,10], among other approaches, to achieve the optimal outcome. This study proposes a novel approach to the optimization problem, employing DOE techniques such as the Taguchi analysis and the FCCCD method. These methods have been developed over time and are readily applicable, combined with the software Minitab. The optimization process is divided into two stages. The initial stage involves identifying the most influential factor(s) affecting the force amplification ratio using the Taguchi method. The results of the Taguchi analysis demonstrate the impact of each factor on the force amplification ratio. As depicted in Figure 5 and Table IV, only two factors, the angle φ_2 and the ratio *BC/CD*, exert a significant influence on the force amplification ratio. The number of variables in (9) can be reduced by assigning specific values to the less influential factors. For instance, the angle α can be assigned the value of 0° to achieve the highest *FA* or it can be selected at 20° to provide a smooth speed profile [10]. This assignment is contingent upon the design intention of the designer. In order to achieve the desired *FA*, it is necessary to identify the optimal combination of φ_2 and *BC/CD*. Consequently, at the second stage, FCCCD is employed to ascertain the dependence of *FA* on these factors. The relationship between *FA* and these factors, which is a nonlinear function, as demonstrated in (9), can be described as a second-order algebraic function (10). This representation facilitates the calculation of *FA* and suggests the potential for developing a program to automate the design process of this mechanism. As shown in the preceding section, when $\varphi_2 = 89°$, $\varphi_3 = 2°$, $\varphi_4 = 2°$, $\alpha = 0°$, and $u = 0.8$, the maximum value of *FA* is 639.0486. This result is considerably higher than those obtained in previous studies, such as *FA*=20.4 [10]. In contrast to other optimization methods, which often yield a single optimal result, the FCCCD method can provide a range of combinations of factors (φ_2 and *BC/CD*) that yield the same required force amplification ratio. This provides the designer with a greater degree of flexibility when attempting to create a five-point double toggle mechanism with a specific force amplification factor.

This study also presents a methodology for calculating the dimensions of the links from the optimal results. To the best of the authors' knowledge, this procedure has not been adequately delineated in any existing references. Once the angles φ_2, φ_3, φ_4 of the mechanism at the mold-closing stage and the geometric parameters α and *BC/CD* have been determined, the designer can readily calculate the dimensions of the links using (11) and other specifications, such as mold height and opening/closing stroke. Once the geometry of the mechanism has been determined, the designer can proceed to the subsequent stages of the design process, such as the dynamic analysis of the clamping unit, design of link shape, or preparation of input parameters for other optimization techniques [9, 10].

V. CONCLUSIONS

In an injection molding machine, a clamping unit provides linear motion to open and close two halves of the mold. Furthermore, the clamping force generated is sufficient to maintain the mold in a closed position throughout the injection molding process, including the injection and packing phases. A five-point double-toggle mechanism is frequently employed in a clamping unit to augment the force generated by a hydraulic cylinder, thereby achieving the requisite clamping force. A high force amplification ratio is a desirable characteristic, as it allows the creation of a high clamping force from a relatively small input force from the cylinder crosshead. The relationship between this ratio and other factors is described as a nonlinear function with multiple variables. To facilitate the optimization of this ratio, the Taguchi method is employed to assess the impact of each variable on the force amplification ratio. Once the most influential variables have been identified, the number of variables for optimization is reduced by fixing the values of the less influential ones. The Taguchi method also helps ascertain the range of factors within which there is a significant variation in the force amplification ratio. Subsequently, the response surface method, Face-Centered Central Composite Design (FCCCD), is employed to construct a second-order regression model, which approximates the original function. By replacing the nonlinear function with a second-order algebraic function, the optimization of the force amplification ratio can be readily achieved. The regression model demonstrates a high degree of correlation with the original function, with an *R*-squared value of 99.01%. This enables the direct calculation of the force amplification ratio. The regression model offers a range of variable combinations that yield the same force amplification ratio, providing flexibility in the design of this type of mechanism. With the specifications of the working conditions of the clamping unit, including mold height, opening/closing stroke, etc., the link dimensions can be calculated. The optimal result, as determined by Minitab, indicates that the maximum force amplification ratio (*FA*=639.0486) can be achieved when φ_2=89°, φ_3=2°, φ_4=2°, α=0°, and u=0.8. The maximum value obtained by this method is considerably higher than those attained in previous research. Furthermore, this study presents a methodology for the design of a five-point double-toggle mechanism, with the objective of ensuring the desired force amplification ratio. The dimensions derived from the proposed method can be used in subsequent stages of the design process, such as the configuration of the link shape, the modeling of the mechanism's dynamics, and the preparation for further optimization problems.

Further work is required to address several aspects. Factors, such as geometric errors in link dimensions due to manufacturing processes and variations in mechanical properties of materials, can affect the distribution of stress and deflections of links under load. This results in variations in the behavior of the upper and lower groups of the mechanism, which may lead to a reduction in the clamping force and reliability of the clamping unit. The optimization of the link shape, which provides suitable stiffness while minimizing weight, is also a topic for further investigation. Consequently, future research will concentrate on the assessment of the reliability of the clamping unit, the topology optimization of link shapes, the geometric modeling, and the dynamic simulation of the mechanism. Furthermore, the fabrication of the clamping unit and the execution of several experiments are necessary for the validation of the mechanism's design.

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