Computational study on the clamping mechanism in the injection molding machine

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Abstract
The clamping mechanism plays an important role in obtaining high-quality products of the injection molding process. The clamping mechanism of the five-point double-toggle has been widely used for the high-speed plastics injection molding machine. The purpose of this paper is to optimize the five-point double-toggle clamping mechanism through multi-body dynamics analysis. This work also provides guidelines and a clear understanding for designing the clamping system in an injection molding machine with various clamping forces. The theoretical calculation has been handled first and then the computational model has been verified in this study. In addition, the effects of clamping forces on the main dimensions, including movable-fixed plate thickness, tie-bar diameter, and average link cross-section have been investigated theoretically and numerically. The results show that the optimal design allows reaching a high force amplified ratio and that the obtained mechanism has good kinematic performance and works steadily with lower energy consumption and lower cost than the preliminary design. Moreover, the relationships between the parameters such as the critical angles of the double-toggle clamping mechanism, the ratio of force amplification, and the stroke of movable mold have been found in this work. The optimized parameters will yield useful knowledge to design and manufacture the clamping mechanism of the microinjection molding machine in practice.

Keywords Injection molding machine · Clamping mechanism · Optimized parameters · Force amplification · Computational study

1 Introduction
Nowadays, the injection molding process has been widely used for the mass production of complicated geometries with high-dimensional precision. Molded part quality largely depends on the accuracy of the injection molding machine and processing parameters as well. The molding clamping system is one of the most crucial systems that directly affect the dimensional accuracy and the quality of the molded products. The clamping unit is used to keep the mold shut against the forces developed when injection pressure pushes melted plastics into the cavity. There have been many studies on the nine-link type mold/die clamping mechanism due to their wide applications in the industry for years. As far as theoretical analysis is concerned, Choubey and Rao approached kinematic errors more precisely by taking into account both minimizing structural and mechanical errors [1]. For the experimental method, there are many studies presented in the literature. For example, Rao et al. [2] and Huang and Lin [3, 4] used strain gauge-based sensors to describe the strain and stress on the four tie bars and to further predict the clamping force. Chang et al. investigated a new design of a tie-bar-less clamping system for microinjection molding machines [5]. The ultrasonic probes are established on the ends of the tie bars, to measure the clamping force based on the proposed model relating the ultrasonic
propagation time to the clamping force of the mechanism are also carried out by Zhao et al. [6]. With regard to theoretical analysis and experimental solution, a case study was conducted by Chang et al. in order to consider kinematic error equations of links and tolerances in nine-link type double-toggle mold/die clamping mechanisms [7]. Lin and Hsiao explored the friction effects at pin joints of the clamping mechanism during real clamping operation [8]. They also improved the clamping system to enhance the performance of stroke ratio and thrust saving as well as design a reasonable space for the ejector unit [9, 10]. In other work, there have been different approaches presented based on theoretical analysis and simulation. Zhang et al. analyzed the performance of the mold clamping mechanism in an injection molding machine by evaluating the microstructure, stress, travel, speed ratio, and the ratio of amplified force. Then, the designed clamping mechanism with microstructure is studied by the theoretical analysis and simulation [11]. The genetic algorithm (GA) numerical technique to optimize the key design parameters of the clamping mechanism is adopted by Huang et al. [12]. Fung et al. also investigated the kinematic and sensitivity of a newly designed toggle mechanism formed by two slider-crank mechanisms [13]. Besides, Li et al. [14] utilized MATLAB to get the optimized parameters of the clamping mechanism such as stroke ratio, velocity ratio, and the amplified force ratio of the microinjection molding machine. Moreover, Jiao et al. studied the clamping characteristics of different types of clamping units thanks to the finite element method of the ABAQUS tool and experiment in order to investigate the clamping uniformity and evaluate the clamping force repeatability precision [15]. In 2006, Roland Felix invented a clamping mechanism for a clamping unit of an injection molding machine where the system can lock the nut on the screw shaft and prevent reverse rotation of the locking nut [16]. Chen et al. reported that Watt-chain double-toggle mold clamping mechanism is better than the multiple-joint double-toggle mold clamping mechanism due to the reduction of acceleration and driving force [17]. In addition, the kinematics of the clamping mechanism was rigorously investigated to reduce acceleration [18].

In those previous works, most of the approaches are limited to theoretical analysis or/and experimental investigation of the clamping mechanism design. It can be seen that the nine-link mechanism should be described by a multi-body dynamics model for a rigorous analysis prior to manufacturing a real injection molding machine to reduce production costs. In addition, up to our knowledge, one cannot find a complete guideline in the literature to help quickly design the clamping mechanism for a wide range of forces in precise microinjection molding machines. In this regard, the objective of this work is to design the five-point double-toggle clamping mechanism of a microinjection molding machine by optimal multi-body dynamics numerical analysis and theoretical verification. The approach of multi-body dynamics simulation has been applied to quickly and easily design the clamping mechanism of microinjection molding machines for a wide range of clamping forces. Besides, the multi-body dynamics simulation could help designers inspect the motion of the mechanism more intuitively. Thanks to the results of this analysis, we could have useful design suggestions and more insight for designing the clamping system in an injection molding machine.

2 Overview of the mechanism

2.1 Problem description

The clamping unit plays an important function in closing, opening, and holding the mold in the injection molding process. Noticeably, it also allows preventing molten plastics from going out from the cavities during injection molding. The clamping mechanism must induce a sufficient force up to approximately hundreds or even thousands of tons to hold the mold closed. In general, the toggle mechanism presents the advantages of motion efficiency and energy saving, and thus it is commonly adopted as the clamping mechanism for injection-molding machines [12]. The five-point double-toggle mold-clamping mechanism is the most extensively used for injection molding machines with clamping force between 50 and 500 metric tons due to the ideal kinematic velocity system and the mechanical features [8]. In recent years, injection molding machines having this clamping mechanism with forces of up to 5000 metric tons have been developed continuously. In this regard, we have selected the five-point double-toggle mold-clamping mechanism for the present investigation. The diagram of transmission and the applied force are shown in Fig. 1.

A five-point double-toggle clamping mechanism is adopted to exploit the generation of large output clamping force with a small input value of crosshead force. There are nine links of the clamping mechanism such as fixing mold plate, two links \(AC, BC, \) and the horizontal line, respectively. The angles \(\alpha_1, \beta_1, \alpha_2, \beta_2\) are formed between the segments \(AC, BC, \) and the horizontal line, respectively, and the angles between the segments \(CD, DE, \) and the vertical line are \(\alpha_1, \beta_1\) respectively. Figure 2 presents the free body diagram of a five-point double-toggle clamping mechanism. The force \(F_c\) induced by the clamping hydraulic cylinder will act on link \(DE\) via the force \(T_c\). The force is transmitted to the link \(ACD\), and it causes the link \(BC\) to rotate. At point C, there are three forces acting on the point C, namely
At point $E$, the total force exerted by the crosshead and the link $DE$ is shown in Fig. 2d, and their balance is expressed in Eq. (3):

$$-F_{34x} + F_{04x} = 0$$

Figure 2e shows the free-body diagram at point $B$ of the link $BC$, where the force balance can be derived in Eq. (4):

$$F_{25x} - F_{05x} = 0$$

By solving Eqs. (2), (3) and (4), the solutions can be expressed as follows: $P = 2T_2 \cos \alpha_2$; $F_C = 2T_3 \sin \beta$; $T_4 = T_3 \cos \beta$; $T_3 = \frac{T_3 \sin \alpha_1 + T_2 \sin \beta}{\cos \alpha_1}$; $T_1 = T_2 \frac{\cos (\alpha_1 + \beta)}{\cos (\alpha_2)}$.

The ratio of force amplification is defined as

$$K = \frac{P}{F_C}$$

Substituting all force values into Eq. (5), the exact expression for $K$ can be obtained.

$$K = \frac{\cos \alpha_2}{\tan \beta \left( \sin \alpha_2 + \sin \alpha_1 \frac{\cos (\alpha_1 + \beta)}{\cos (\alpha_2)} \right)}$$

### 2.2 Preliminary design

Figure 3 presents a kinematics diagram of the typical five-point toggle-type clamping unit in the injection molding machine, where the system is in a state corresponding to mold closure (the above figure) and mold opening at a maximal stroke (the below figure). At the initial design step, the length ratio of the link $AC$ to link $CB$ is first given the value of 0.77. Applying the geometrical analysis of the clamping mechanism at mold-closing and mold-opening states, the approximately total lengths according to the horizontal direction of the links are obtained by Eqs. (7) and (8)
respectively. Therefore, the stroke of the movable mold is 150 mm, which can be expressed by $L_2 - L_1$. The distance between the top link and bottom link can be described by Eq. (9), and the stroke of the hydraulic cylinder and the $IA$ length can be defined by Eqs. (10) and (11), respectively.

$$L_2 = AC \cos \alpha_1 + BC \cos \alpha_2 = BC(0.77 \cos \alpha_1 + \cos \alpha_2) \quad (7)$$

$$L_1 \approx \sqrt{BC^2 - AC^2} = 0.64BC \quad (8)$$

$$\frac{b}{2} = EG + ED \cos (\beta - \alpha_3) + CD \cos \alpha_3 + AC \sin \alpha_1 \quad (9)$$

$$L_0 = IA + AC \cos \alpha_1 - CD \sin \alpha_3 - DE \sin (\beta - \alpha_3) \quad (10)$$

The preliminary parameters are taken into account such as the initial angles $\alpha_1 = 4^\circ; \alpha_2 = 0.77 \alpha_1 = 3.07^\circ; \beta = 15.18^\circ$, the crosshead length of 80 mm; $EG = 40$ mm, $CD = 42$ mm, and $b = 286$ mm ($b > 2AC$). Substituting the values of the initial angles into Eq. (6), the ratio of force amplification is calculated as $K \approx 29.7$. By solving Eqs. (7), (8), (9), (10) and (11), the lengths of the links are obtained: $BC = 150$ mm, $AC = 115$ mm, $ED = 55$ mm, $L_0 = 170$ mm; and $IA = 82$ mm. To ensure enough space for connecting the rod of the cylinder to the crosshead, we chose eventually $L_0 = 200$ mm, and $IA = 112$ mm. From the parameters above, a full 3D model of the clamping mechanism system has been designed as shown in Fig. 4.

### 3 Computational model

A simulation procedure is described first for the computational study of the clamping mechanism dynamics. Validation of the simulation approach will be presented before moving on to the present design. In the current design, preliminary design analysis is handled first, and an optimal design model is performed later. In addition, a setup of boundary conditions and mesh convergence tests are provided for cross-section dimension calculation purposes.

#### 3.1 Establishing boundary conditions of modeling simulation

Nowadays, with the strong development of computer sciences, the finite element method is known as one of the most helpful numerical techniques used in the design process thanks to complex calculation and detailed analysis with improved accuracy and reduced computing time [11, 15]. In the present study, the software study edition of HyperWorks with modules of MotionView and HyperStudy were simultaneously used to simulate the clamping mechanism and perform its optimization under multibody dynamics.

As far as the tool MotionView is applied, the procedure can be described by the following steps. Firstly, the linking points of the structure were created by providing the coordinate points. The latter is obtained from 3D CAD software and is used to design the initial structure. Secondly, the 3D data was imported in order to build the links, and then applying the materials for these parts was set up. Thirdly, the joints of the clamping system were produced to connect the different links. Then, the imposed motions and driving force were applied as boundary and loading conditions. A cylinder
drives the mechanism’s motion, with the angular speed of the crank toggle being one rad/s. Two damping springs were set up at the contact mold position and crosshead to measure the force of the movable mold and crosshead. This force simulation method was also adopted to describe the clamping force before [11]. Finally, the output parameters such as forces in the links and angles were established before moving on to the run analysis.

In this study, the multi-body dynamics simulation results of the clamping mechanism have been validated by Huang et al. model [12]. The parameters of the clamping mechanism include the clamping force of 200 tons, the stroke of the movable mold of 470 mm, the length of links $AC$, $BC$, $CD$, $DE$, and other dimensions according to the given model [12]. The results of movable mold motion using the simulation technique, including speed and ratio of speed amplification between the movable mold and crosshead, are shown in Fig. 5a, b, respectively. Figure 6a, b show the force on the crosshead and the ratio of force amplification of the optimal method, respectively. The present simulation technique’s results agree with the optimal results presented by [12], especially the ratio of speed amplification curves of the two optimal methods are completely superposition. The forces measurement has been defined thanks to the numerical analysis approach, and the validation has been carried out as shown in Fig. 6a, b. Crosshead forces and the ratio of force amplification verified show correction of the simulation.

3.2 Preliminary dynamics analysis

Based on the modeling simulation established in Sect. 3.1. In this section, a clamping mechanism model with parameters of preliminary design is set. With the clamping force of the mechanism being 30 tons and the stroke of movable mold being 150 mm, the motion of the structure is set with an angular speed of link $ACD$ (crank link) of one rad/s, and
the motion finish when the value of $\alpha_1$ angle (crank angle) reaches the critical value in terms of angular position.

The results of the initial dynamics simulation of clamping force and crosshead force and the ratio of force amplification are shown in Fig. 7a, b, respectively. It can be seen that the maximum clamping force of 30 tons, which is the maximum force qualified of the mechanism, the maximum force, and the force when the value of $\alpha_1$ angle reaches the critical value on the crosshead force are around 16,274 N and 9183 N, respectively. The ratio of force amplification is around 32.6, which shows a good agreement with the theoretical calculation of 29.7. This result is relatively accurate, and it can thus be established that the boundary conditions and simulation data set are reasonable. The ratio of force amplification largely depends on the angles $\alpha_1$, $\alpha_2$, $\alpha_3$, and $\beta$, which means that the clamping force begins to strongly increase when the mold plates contact each other. The smaller values of these angles lead to the larger values of the ratio of force amplification in the clamping mechanism.

3.3 Establishing optimized modeling on HyperStudy

Optimizing the key parameters of the clamping unit such as the length of links, the initial angle’s values $\alpha_1$, $\alpha_2$, $\alpha_3$, and $\beta$ is extremely important because this allows designing the toggle mechanism with efficient motion and energy saving when operated [12, 14, 15]. In this study, a new solution by combining the modules of MotionView for dynamics simulation and HyperStudy for digital graph analysis has been applied in order to find the optimized parameters of the clamping mechanism.

For the computational model on HyperStudy, the coordinate equations are employed to define the relative position of
these variables. According to the geometric description shown in Fig. 3, the coordinate equations at points A, B, C, D, and E can be expressed by Eqs. (12), (13), (14), (15) and (16), respectively.

\[
\begin{align*}
\begin{cases}
x_A &= IA \\
y_A &= \frac{b}{2}
\end{cases} \quad (12)
\end{align*}
\]

\[
\begin{align*}
\begin{cases}
x_B &= IA + AC \cos \alpha_1 + BC \cos \alpha_2 \\
y_B &= \frac{b}{2} - AC \sin \alpha_1 + BC \sin \alpha_2
\end{cases} \quad (13)
\end{align*}
\]

\[
\begin{align*}
\begin{cases}
x_C &= IA + AC \cos \alpha_1 \\
y_C &= \frac{b}{2} - AC \sin \alpha_1
\end{cases} \quad (14)
\end{align*}
\]

\[
\begin{align*}
\begin{cases}
x_D &= IA + AC \cos \alpha_1 - CD \sin \alpha_3 \\
y_D &= \frac{b}{2} - AC \sin \alpha_1 - CD \cos \alpha_3
\end{cases} \quad (15)
\end{align*}
\]

\[
\begin{align*}
\begin{cases}
x_E &= IA + AC \cos \alpha_1 - CD \sin \alpha_3 - DE \sin \beta \\
y_E &= \frac{b}{2} - AC \sin \alpha_1 - CD \cos \alpha_3 - DE \cos \beta
\end{cases} \quad (16)
\end{align*}
\]

In the computational model, the two points A and B are fixed, and there are four variables used in the mold closing state resolution, including the length of CD, the length of DE, the angle \( \alpha_3 \), and the angle \( \beta \). In this work, these variables are limited with the upper and lower values selected as \( 32 \leq CD \leq 46 \) (mm), \( 35 \leq DE \leq 55 \) (mm), \( 0 \leq \alpha_3 \leq 16 \) (°), and \( 0 \leq \beta \leq 16 \) (°). The travel distance of the movable mold plate used is 150 mm. The optimal angles \( \alpha_3 \) and \( \alpha_2 \) and the length of AC and BC links are defined through the position of point C. Finally, the problem is solved simultaneously by MotionView and HyperStudy integration. In HyperStudy, the data are taken from the dynamics simulation results on MotionView, and then it is analyzed according to the iteration statement.

### 3.4 Cross-section of the clamping mechanism according to the optimal results

The main dimensions of the clamping unit were introduced in the present study such as the cross-section of links, the diameter of the tie bars and the thickness of the plates. The links are subjected to compressive forces when the system is operated, so its strength condition is calculated by Eq. (17), where \([\sigma_n]\) is admissible compressive stress of the material, \(A\) is the cross-section area of link, and \(T\) is the internal force value of the link. It can be realized that the link BC has the maximum internal force \(T_2\), so this link is taken into account for strength analysis. As far as tie bars are concerned in the closing mold state, they are affected by a tension force, the equation for strength condition is obtained by Eq. (18), where \([\sigma_s]\) is the admissible tensile stress of materials; \(A_t\) is the cross-sections area of tie bar. In addition, when the mechanism reaches the maximum clamping force, the moment of deflection caused on the movable mold plate is the largest. Therefore, the thickness of all plates will be computed according to the dimension of the movable mold plate. The size of the movable mold plate is 350×350 mm, and the hole at the plate center is 60 mm. The strength condition of the plate is defined by Eq. (19), where \([\sigma_s]\) is the admissible tensile strength of the material, \(W_t\) is section modulus, \(M_{\text{max}}\) is the maximal moment of deflection on the plate and the cross section of the plate is \(b\times h\), in which \(b\) and \(h\) are the width and the thickness of plate, respectively.

\[
\Sigma A_{\text{cross links}} \geq \frac{\Sigma T}{[\sigma_n]} \quad (17)
\]

\[
\Sigma A_{\text{tie bar}} \geq \frac{P}{[\sigma_s]} \quad (18)
\]

\[
W_t \geq \frac{M_{\text{max}}}{[\sigma_s]}, \quad W_t = \frac{bh^2}{6} \quad (19)
\]

### 3.5 Mesh convergence tests

A mesh study is carried out with the computational domain of plate shape in the present study. The beam has a cross section of 150×30 mm and a length of 300 mm, and two supports are applied at the ends of the beam. In addition, a distributed load of 120,000 N over 240 mm length is applied on the beam as shown in Fig. 8. The computational domain discretization was chosen to be done with 5×5×5 uniform hexahedral elements after a preliminary mesh convergence study by using eight types of mesh, namely 15×15×15, 10×10×10, 6×6×6, 5×5×5, 3×3×3, 2×2×2, 1.5×1.5×1.5, and 1×1×1. In the mesh convergence test, the simulation results have been examined in terms of stress, and the solution with the mesh of 5×5×5 is found to be reliable in providing an almost converged solution with a reasonable computation time under available resources. The mesh ratio is defined by the ratio of mesh size to beam thickness as shown in Fig. 9.
3.6 FEM model for durability simulation of the clamping mechanism

In this study, the software HyperWorks [20] with modules of Hypermesh and Optistruct is used for durability simulation. All components of the clamping mechanism are set at the closing mold state. After meshing, the connections between the parts such as bolt and nut, rotary pin, and welding joint are created by connectors.

As boundary conditions, the impact force on the movable mold and the reaction force on the fixing mold taken are both equal to 30 tons. The force applied to the molds is distributed over the entire contact area between the movable and fixing molds. The problem is solved with the linear static state of the Inertia Relief Analysis module in Optistruct. When solving the static problem, finite element solvers will handle the Eq. (20), where $[M]$ is the global stiffness matrix, $x$ is the displacement vector response, and the external forces vector applied to the structure are defined as $f$.

$$f = [M]x$$

(20)

4 Results and discussions

4.1 The optimized designing clamping unit

With the computational model set up above, the value of the optimal parameters obtained by simulation is as follows: $AC = 106$ mm, $BC = 119$ mm, $\alpha_1 = 3.04^\circ$; $\alpha_2 = 2.76^\circ$; $\alpha_3 = 13.8^\circ$; $\beta = 4.2^\circ$; $CD = 45$ mm; $DE = 50$ mm; $EG = 43.65$ mm, and the ratio of force amplification is 129.4, where the input parameters included the maximal clamping force of 30 tons, the plate thickness of 30 mm, the cross-section area of links of 9 cm², and the tie-bar diameter of 28 mm. The description of the clamping force and the ratio of force amplification as a function of crank angle ($\alpha_f$) is plotted in Fig. 10a, b. It can be noticed that the crosshead force is much smaller after the optimization and the clamping force increase when the value of the crank angle decreases. The increase up to 30 tons occurs as the movable mold starts to contact the fixed mold. The optimal value of the ratio of force amplification is four-time larger than that of the preliminary design.

Figure 11 shows the ratio of speed amplification ($K_v$) between the movable mold and crosshead as a function of the crank angle, where $K_{v_{\text{max}}}$ and $K_{v_{\text{min}}}$ represent the local maximum and local minimum in the $K_v$ curve. According to the proposed model, the ratio $K_{v_{\text{max}}}/K_{v_{\text{min}}}$ represents the smoothness of the motion of the movable mold [12]. In Fig. 11, the ratio $K_{v_{\text{max}}}/K_{v_{\text{min}}}$ of the optimal model is 1.15; that value is smaller than the preliminary design is 1.298. It can be seen that the optimal design has the motion of movable mold smoother and good kinematic performance than the preliminary design.

The stroke of the crosshead of the preliminary and optimal design as a function of time is shown in Fig. 12. Here, the stroke ratio between the movable mold and crosshead is $K_d$, which represents the efficiency of the mechanism. The ratio of the stroke of movable mold to the stroke of crosshead is as large as possible [12]. In Fig. 12, with the stroke of the movable mold of 150 mm, the value of $K_d_{\text{preliminary}}$ of preliminary design is 0.86 smaller than the value of $K_d_{\text{optimal}}$ of optimal design of 0.886, which means the stroke of the crosshead of optimal design is shorter than preliminary design, through improves the efficiency of the mechanism. In other words, the optimal design works steadily with lower energy consumption and lower cost than the preliminary design.
Figure 13 describes the effects of the link angle according to time. In general, the optimal angle values are smaller than the preliminary angle ones. However, the beta angle shows a big difference between the optimal value and the preliminary one. The critical angle values match well with the previous studies handled by Matlab simulation [14, 19], with the alpha critical values of 3.32° and 3.193°, respectively.

Figure 14 shows the relationship between the link angles and the ratio of force amplification. The clamping force strongly increases when the angles move toward the critical values. The simulation results show a very good agreement with the theoretical calculation. In addition, the angles alpha 1 and alpha 2 represent a significant impact on the ratio of force amplification in comparison with the beta angle. It can be concluded that determining the optimal critical angle will create a large clamping force and this is known as an advantage for the toggle clamping mechanism.

The relationship between the travel distance and the toggle mechanism angles is presented in Fig. 15. The graph shows that the effects of angle beta on the stroke of the movable mold plate are much smaller than that of alpha 1 and alpha 2. Furthermore, when the angles move to the critical values, the rate of movement is slower. This indicates that the optimal mechanism system works smoothly and stably.

4.2 The testing durability of the clamping unit

The relationship between the clamping forces ranging from 10 to 50 tons and the main dimensions such as the movable mold plate thickness, the diameter of the tie bar, and the average cross section of the links are shown in Fig. 16.
stresses used for the theoretical calculation and numerical simulation in comparing with admissible stresses is around 2.5% difference, and the permeable safety factor is not introduced for the study. Obviously, the main dimension increases with the clamping forces. The differences between the theoretical calculation and simulation of the movable mold plate thickness, the tie-bar diameter, and the average links cross-section are 4.4%, 1.7%, and 11.8%, respectively. In general, the clamping forces are linearly proportional to the average links cross-sections, whereas they have been found to be nonlinear with the movable mold plate thickness and the tie-bar diameter for smaller clamping forces. The detailed values of the main dimensions corresponding to the clamping forces are presented in Fig. 16.

To have a visual insight, Figs. 17, 18, 19 and 20 provide simulation results of stress, which occurs at key joints of the clamping mechanism system at the closing mold state. In this study, the main components are designed with safety factors ranging from 1.13 to 1.77. Figure 17 entirely visualizes the stresses on the clamping unit where maximum stress of 336.6 MPa happens at the junctions between the plate and the links. Figure 18 shows a clear visualization of the maximum stress of 328.7 MPa on the tie bar where it connects to the fixing plate. Additionally, the maximum stress of 249.4 MPa on the links occurs near the joint.
with the movable mold plate as shown in Fig. 19. Regarding the movable mold plate, the maximum stress of 293.1 MPa appears around the hole as indicated in Fig. 20.

### 4.3 The prototype fabrication of injection molding machine and molded product

A complete 3D design model of the injection molding machine has been built as shown in Fig. 21a, where the clamping system is used with a force of 30 tons. An injection molding machine also has been manufactured successfully with the parameters found above as shown in Fig. 21b. Initially, the machine worked well during the testing process, and some of its characteristics have been measured to assess its performance such as the stiffness of the structure, its kinematic and stability. In addition, Fig. 22 depicts a prototype article of a TIR optical element molded by the injection molding machine designed in the present study.
Fig. 16  Relationship between the clamping force and main dimensions of the clamping unit

![Graph showing the relationship between clamping force and dimensions](image)

Fig. 17  The stress on the clamping unit in the closing mold state

![Contour plot showing stress distribution](image)

Fig. 18  The stress on the tie bar of the clamping unit in the closing mold state

![Contour plot showing stress distribution](image)
**Fig. 19** The stress on links of the clamping unit in the closing mold state.

**Fig. 20** The stress on the movable plate of the clamping unit in the closing mold state.

**Fig. 21** The full 3D design of the machine (a) and the injection molding machine fully manufactured (b).
is much larger than that of the beta when the angle tends to zero. Secondly, the cross-section dimension of the link is proportional to the clamping forces. Furthermore, there are many simulation results, which show the maximum stresses occurring at the movable-fixed plate, the tie-bar, and the links.

**Nomenclatures** (0): Fixed mold plate; (1): Link ACD; (2): Link BC; (3): Link DE; (4): Crosshead of the piston; (5): Movable mold plate jointed to the link BC; AC: Length of link ACD; CD: CD length of link ACD; IA: The length of the part connecting the ACD link to the fixing plate; BC: The length of the BC link; EG: The half-length of the link crosshead; α1: Crank angle (the angle between AC and horizontal axis at closing mold); α2: The angle between BC and horizontal axis at closing mold; α3: The angle between CD and vertical axis at closing mold; β: The angle between DE and vertical axis at closing mold; $F_c$: The force of the hydraulic cylinder; $P$: The clamping force of the machine; $F_{ij}$: The force induced from link i acting on link j with the horizontal direction; $F_{ijy}$: The force induced from link i acting on link j with the vertical direction; $b$: The vertical distance between two supports IA; $L$: The stroke of the movable mold plate; $L_0$: The stroke of the piston; $L_c$: The length of a toggle at closing mold; $T_0$: The length of a toggle at opening the mold; $T_1$: The force on link ACD with AC direction; $T_2$: The force on link ACD with CD direction; $T_3$: The force on the link BC; $T_4$: The force on the link DE; $K$: The ratio of force amplification; $K_v$: The ratio of speed amplification (between the movable mold and crosshead); $K_{max}$: The ratio of speed amplification at the local maximum; $K_{min}$: The ratio of speed amplification at the local minimum; $K_v$: The ratio of stroke amplification (between the movable mold and crosshead); $K_{\text{optimal}}$: The ratio of stroke amplification of optimal design; $K_{v,\text{preliminary}}$: The ratio of stroke amplification of preliminary design

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**Declarations**

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**Author contribution** Van-Duong Le performed the calculation and the simulation data and drafted the manuscript. Van-Thanh Hoang contributed to the design, implementation of the research, and to the writing of the manuscript. Quang-Bang Tao contributed to the analysis of the results and to writing of the manuscript. Lahouari Benabou contributed to the rewriting of the manuscript and aided in interpreting the results. Ngoc-Hai Tran and Duc-Binh Luu processed the numerical analysis data. Jang Min Park contributed to the reanalysis of the results and to the rewriting of the manuscript. Van-Duong Le and Van-Thanh Hoang contributed equally to this work. All authors discussed the results and commented on the manuscript.

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**Author contribution** Van-Duong Le performed the calculation and the simulation data and drafted the manuscript. Van-Thanh Hoang contributed to the design, implementation of the research, and to the writing of the manuscript. Quang-Bang Tao contributed to the analysis of the results and to writing of the manuscript. Lahouari Benabou contributed to the rewriting of the manuscript and aided in interpreting the results. Ngoc-Hai Tran and Duc-Binh Luu processed the numerical analysis data. Jang Min Park contributed to the reanalysis of the results and to the rewriting of the manuscript. Van-Duong Le and Van-Thanh Hoang contributed equally to this work. All authors discussed the results and commented on the manuscript.

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The prototype shows a good appearance and a quantitative estimation will be inspected for future work.

**5 Conclusion**

This study has presented the optimal design of the five-point double-toggle mold clamping mechanism in the injection molding machine thanks to theoretical and numerical analysis. The effects of the critical angles of the double-toggle clamping mechanism, the ratio of force amplification, and the stroke of the movable mold have been introduced in this paper. Notably, the optimal value of the ratio of force amplification is four-time larger than that of the preliminary design. The distance between max speed and min speed of optimal design was minimized compared to the preliminary design. This process helps the speed of the movable platen become smoother as stated in Huang et al. study [12]. The stroke ratio between the movable mold and crosshead of the optimal value is 3% larger than that of the preliminary design. In addition, the design parameters, including the movable-fixed plate thickness, the tie-bar diameter, and the average link cross-section depending on various clamping forces are taken into account. The theoretical calculation has been found to agree well with the simulation results. Meanwhile, there are some other findings from the present study. Firstly, the influence of alpha 1 and alpha 2 on the ratio of force amplification and the stroke of the movable mold.
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