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Optimization Design of the Toggle Clamping System for a Plastic Injection Molding Machine

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Keywords： injection molding machine, toggle clamping system, optimization design.

ABSTRACT

The main purpose of this work is to design the toggle system for an injection molding machine (IMM). Its stroke, clamping force amplitude, and close mold speed of the system are taken as the three objective functions by using optimization method and 3D design simulation software. From our numerical results the optimal design are really better than the original design: (1)The relation stroke has been increased. (2)The clamping force amplitude has been improved. (3)The starting velocity of moving platen in the initial point is reduced, and reaches a higher velocity in the middle stage.

INTRODUCTION

Over the years, the emphasis on improving molded product quality characteristics or manufacturing productivity indices of the injections molding machine has resulted in a significant focus by both the industrial suppliers and academic/research institutions (Peterson 1994). In the early days there were methods of drawing the mechanism models and adjusting the linkage dimensions manually. These methods all estimate the mechanism parameters based on the stroke and self-lock demands, however, none of them can optimize the plastic injection molding machine (Liao etc. 2005). The toggle clamp has become one of the most common forms of clamps used throughout industry nowadays. They are available commercially in different type models and satisfy almost all ordinary mold clamping requirements (Tso 1998).

The mechanical amplitude of the toggle clamping mechanism can be as high as 20 ~ 25. It means a large clamp force can be achieved by a relatively small force from the driving cylinder. There are two main advantages for a toggle clamping mechanism. First, the economy of running a much smaller hydraulic cylinder and second, it has a very high spring rate (Cappella 1995). Improper clamping force will result in poor production process with frustrating regularity and at the expense of mold wear, machine and mold damage. The consequences can be crushed vents and gates, hobble molds surfaces, or cocked molds distorted out of square (Rideout 1985, Lin and Hsiao 2003).

An analysis of motion and mechanical characteristics for toggle clamping mechanism of IMM is made by the structural method. The formulas for calculating of the stroke ratio of moving platen and cylinder, the closing amplitude force and the speed variation coefficient are derived to establish the mathematical model for the optimized design (Feng and Cen 2002). By means of the application of the optimum analysis (Arora 1989), 3D design and simulation software, the main purpose of this work is to improve the design of toggle clamping system in the plastic injection molding machines for producers and customers.

SOFTWARES ON THIS STUDY

Three software, MOST, Solidworks, and SimWise 4D, are used in this study to accomplish the works on optimization, mechanical design and mechanism analysis.

Program MOST

MOST, which was developed in C language and used in the batch or interactive mode of computation, is a well-known software for Multifunctional Optimization System Tool. In MOST, the Sequential Quadratic Programming (SQP) is one of the most successful methods for the numerical solution of constrained nonlinear optimization problems. Its modified branch-and-bound algorithm can convert discontinuous design space into a continuous one by dropping discontinuous restrictions and used in solving discrete optimization (Tseng 1996).

Solidworks

Solidworks is a 3D mechanical CAD software.

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It is a parasolid-based modeler, and utilizes a parametric feature-based approach to creating models and assemblies. Its solutions can cover all aspects of the product development process with a seamless, integrated workflow. Designers and engineers can span multiple disciplines with ease, shortening the design cycle, increasing productivity and delivering innovative products to market faster.

SimWise 4D

SimWise 4D is for design and engineering professionals developing products involving assemblies of 3D parts. By simulating the assemblies in its unique virtual environment, one can produce more creative, robust designs and reduce cycle time. It includes components for design validation, including motion and dynamic, stress and deflection, vibration and buckling response, and heat transfer analysis. Using these features at early design stages in the design cycle allows designers to generate alternative designs based on user defined optimization criteria and to eliminate unnecessary design cycle iterations more flexibly.

OPTIMA DESIGN OF THE TOGGLE CLAMPING SYSTEM

The main purpose of this work is to improve the toggle clamping system that will create a benefit for the plastic injection molding machine builders. There are three cost functions in this study: relation stroke, clamping force amplitude, and close mold speed. The decision, on which cost function will be utilized, depends on the builder and customer, for instance:

(1) Even reducing the hydraulic cylinder stroke and increases the moving platen stroke
(2) Increases the clamping force amplitude
(3) Higher close mold speed to save dry cycle time

System parameters

Shown in Figure 1 is the toggle clamping mechanism with six-links whose input and output linkages are two slides, corresponding to the clamping cylinder and moving platen, respectively, for an injection molding machine. Figure 2 shows its mold closing and opening positions. All of the parameters of the toggle clamping system are defined in the following (Dai and Lin 1993):

L_1, L_2, L_3, L_4, L_5= length of every linkage (mm),
γ = angle between L_1 and L_5 (degree),
ω = angle between L_1 and L_3 (degree),
α = an angle to represent the rotation of the triangle linkage (degree),
β = angle between L_2 and the horizontal line (degree),
ϕ = angle between L_4 and the horizontal line (degree),
θ = angle between the horizontal and L_1 in close mold position (degree),
S_m = moving platen stroke (mm),
S_c = cylinder stroke (mm),
Ed = distance between the cross head and the point D (mm),
where D is the fix pivot of this toggle mechanism,
dB = diameter of the point B (mm),
dE = diameter of the point E (mm),
H = distance between the center line of this toggle mechanism and point D (mm).

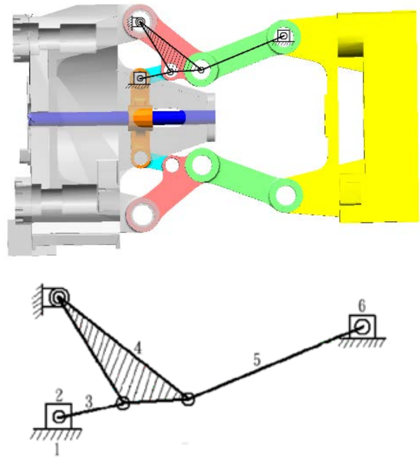

Fig. 1. A skeleton representation of a toggle clamping system.

When moving platen is in the close mold position, L_1 and L_2 are collinear as a line (shown as the dash line in the lower half of Fig 2). The angle between this line and L_1 is defined as α. When moving platen is at the close mold position (upper half of Fig 2), we have $\alpha = 0$, and denote $\beta = \beta_0$ and $\phi = \phi_0$. When moving platen is at the open mold position (lower half of Fig 2), we set $\alpha = \alpha_{max}$, $\beta = \beta_{max}$, and $\phi = \phi_{max}$.

From Fig 2, the moving platen stroke S_m and cylinder stroke S_c can be obtained as:

$$S_m = (L_1 + L_2)cos\theta - L_1 cos(\alpha_{max} + \theta) - \sqrt{L_2^2 - [L_1 sin(\theta + \alpha_{max}) - (L_1 + L_2)sin\theta]^2} \quad (1)$$

$$S_c = L_5[\cos(\gamma + \theta) - cos(\alpha_{max} + \gamma + \theta)] - \sqrt{L_4^2 - [Ed - L_5 sin(\theta + \gamma)]^2} + \sqrt{L_4^2 - [Ed - L_5 sin(\alpha_{max} + \theta + \gamma)]^2} \quad (2)$$

Also, ϕ, β, ω, L_3 can be obtained as:

$$\phi = sin^{-1}\left(\frac{Ed - L_5 sin(\alpha + \theta + \gamma)}{L_4}\right) \quad (3)$$

$$\beta = sin^{-1}\left(\frac{L_1 sin(\alpha + \theta) - (L_1 + L_2)sin\theta}{L_2}\right) \quad (4)$$

$$\omega = sin^{-1}\left(\frac{L_5 \sin\gamma}{\sqrt{(L_5 \sin\gamma)^2 + (L_1 - L_5 \cos\gamma)^2}}\right) \quad (5)$$

$$L_3 = \sqrt{(L_5 \sin\gamma)^2 + (L_1 - L_5 \cos\gamma)^2} \quad (6)$$

Consequently, β_0 and ϕ_0 can be obtained by giving $\alpha = 0$, to have:

$$\phi_0 = sin^{-1}(\frac{Ed - L_5\,sin(\theta+\gamma)}{L_4}) \quad (7)$$

$$\beta_0 = sin^{-1}(\frac{L_1\,sin\,\theta - (L_1+L_2)sin\theta}{L_2}) \quad (8)$$

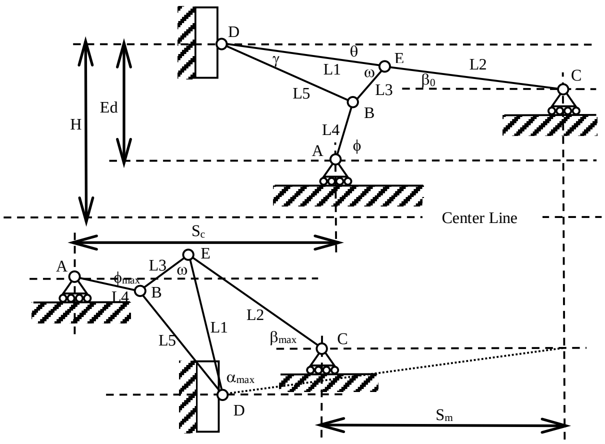

Fig. 2. A skeleton representation of a toggle clamping system.

Also, β_{max} and ϕ_{max} can be obtained by giving $\alpha = \alpha_{max}$, to have:

$$\phi_{max} = sin^{-1}(\frac{Ed - L_5\,sin(\alpha_{max}+\theta+\gamma)}{L_4}) \quad (9)$$

$$\beta_{max} = sin^{-1}(\frac{L_1\,sin(\alpha_{max}+\theta) - (L_1+L_2)sin\theta}{L_2}) \quad (10)$$

Design variables and constrains

Considering the design of the whole toggle clamping system, there are seven variables assigned in this study as the design variables, yielding the design variable vector is given by:

$$X = [L_1\; L_2\; L_4\; L_5\; \gamma\; \theta\; \alpha_{max}]$$

Referring to the reference (Dai and Lin 1993) and the experiences from the designers of the cooperative who design IMM for many years, there are totally twenty constraints needed in this study:

(1) Prevents the over-locking: According to friction circle theory, the outset angle should not be bigger than 150. Therefore, one has the following first constraint:

$g[1] \equiv \alpha_{max} + \theta + \phi_{max} + \gamma - 150 < 0$ (11)

(2) Slanting platen angle restriction is the second constraint:

$g[2] \equiv \beta_{max} - 50 < 0$ (12)

(3) Critical θ angle restriction is in the range of $4° < \theta < 6°$. It should be rewritten as two constraints:

$g[3] \equiv \theta - 6 < 0$ (13)

$g[4] \equiv 4 - \theta < 0$ (14)

(4) From the shape of triangle linkage shown in Figure 3, it is obviously that the following constraint is necessary:

$g[5] \equiv \frac{dB}{2} + \frac{dE}{2} - L_3 < 0$ (15)

(5) The length of linkage L_1 and L_5 have to be restricted to avoid collision with another linkage when they passing through the vertical position (Figure 4):

$g[6] \equiv L_1 + \frac{dE}{2} - H < 0$ (16)

$g[7] \equiv L_5 + \frac{dB}{2} - H < 0$ (17)

(6) From the design experiences, there is a suitable ratio between L_1 and L_2:

$g[8] \equiv L_1 - 0.9 \times L_2 < 0$ (18)

$g[9] \equiv 0.686 \times L_2 - L_1 < 0$ (19)

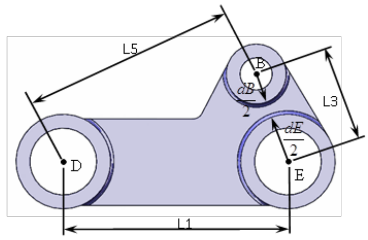

Fig. 3. Shape and dimensions of triangle linkage

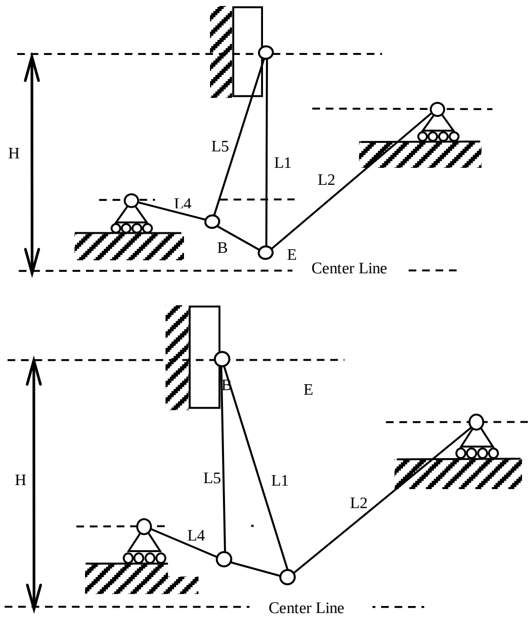

Fig. 4. The positions when L_1 and L_5 are vertical

(7) Outset angle α_{max} restriction listed in the following is another constraint:

$g[10] \equiv 90 - \alpha_{max} < 0$ (20)

(8) Angle γ restriction given in the following is the eleventh constraint:

$g[11] \equiv 12.5 - \gamma < 0$ (21)

(9) Outset angle ϕ_0 restriction ($75° < \phi_0 < 85°$) defines another constraint. It is rewritten to be two constraints:

$g[12] \equiv 75 - \phi_0 < 0$ (22)

$g[13] \equiv \phi_0 - 85 < 0$ (23)

(10) Outset angle ω restriction (in this study is $70° < \omega < 90°$) provides two constraints:

$g[14] \equiv 70 - \omega < 0$ (24)

$g[15] \equiv \omega - 90 < 0$ (25)

Furthermore, from the definitions of the angles ϕ, β, ω in equations (3) ~ (5), the following constraints should be added to avoid domain errors:

$$g[16] \equiv \left|\frac{Ed-L_5\sin(\theta+\gamma)}{L_4}\right| - 1 < 0 \quad (26)$$

$$g[17] \equiv \left|\frac{Ed-L_5\sin(\alpha_{max}+\theta+\gamma)}{L_4}\right| - 1 < 0 \quad (27)$$

$$g[18] \equiv \left|\frac{L_1\sin\theta-(L_1+L_2)sin\theta}{L_2}\right| - 1 < 0 \quad (28)$$

$$g[19] \equiv \left|\frac{L_1\sin(\alpha_{max}+\theta)-(L_1+L_2)sin\theta}{L_2}\right| - 1 < 0 \quad (29)$$

$$g[20] \equiv \left|\frac{L_5\sin\gamma}{\sqrt{(L_5 sin\gamma)^2+(L_1-L_5 cos\gamma)^2}}\right| - 1 < 0 \quad (30)$$

OBJECTIVE FUNCTIONS

First optimization: optimization of relation stroke

An injection molding machine with longer moving platen stroke indicates it can produce a longer plastic product. The moving platen stroke, therefore, is one of the important specifications for such machines. In the first consideration, the requirement for optimization design is to reduce the hydraulic cylinder stroke and to increase the moving platen stroke by defining the following cost function:

$$F = \left(\frac{S_c}{S_m}\right) \quad (31)$$

Where F represents the relation stroke, a relation between the displaced distances within the cylinder stroke S_c and the moving platen stroke S_m, given in equations (1) and (2). If it is minimized, it means the cylinder stroke is minimized and the moving platen stroke is the maximum. All the lower and upper bounds established in the Table 1, which are given by an IMM producer, are fed to software MOST for obtaining the expected solutions. One then can obtain the hydraulic cylinder stroke, when the design variables, all constraints and cost function of Eq. (31) are also given to MOST.

Table 1. Lower and upper bounds of design variables

	Original Design	Lower bound	Upper bound
L_1(mm)	190	50	220
L_2(mm)	242	190	300
L_4(mm)	66.5	30	90
L_5(mm)	181.47	100	220
γ(degree)	23	5	35
θ(degree)	4.6	4	6
α_{max}(degree)	110	75	120

The initial design of this study comes from a cooperative IMM producer. The original design dimensions, therefore, were obtained from one of their commercial machine. This make the research results more valuable for industrial applications.

Second optimization: optimization of clamping force amplitude

The clamping force of the IMM comes from the restoring force of the tie bars when they were elongated by the clamping system as the curve P_t shown in Figure 5 (Beijing 1979). It is a parabola function of α. Curves of moving plate force P_m and cylinder force P_c are also shown in this figure. They are the output and input force of the toggle system respectively. The intersection of curve P_t and α axis is defined as $\alpha = \alpha_0$. At this position, the moving platen just touches the fixed platen and tie bars just ready to be elongated. The clamping force at this time, therefore, equals to zero. The equations of curve P_t, which is a second order function of α, is given as:

$$P_t(\alpha) = \frac{L_1(1+\lambda)\times C}{2} \times cos\theta \times (\alpha_0^2 - \alpha^2) \quad (32)$$

There are one P_c curve, one P_t curve and three P_m curves numbered P_{m-1}, P_{m-2}, P_{m-3} respectively shown in Fig 5. The cylinder output force is amplified by toggle system to pull tie bars to stretch. The restoring force of tie bars then acts as clamping force to clamp mold to prevent mold opened by injection pressure. Different toggle system designs result to different curves as these three P_m curves. A higher P_m curve means the force amplitude of this toggle system is higher. In other words, under the same cylinder, a higher P_m curve means the toggle system can create larger clamping force. That is our objective function in the second optimization case.

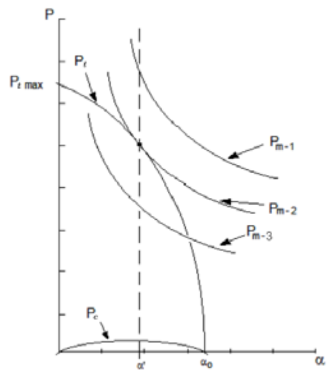

Fig. 5. Curves of P_t, P_m, P_c vs α

From Fig 5 we can find there is a maximum value exist for P_c curve when $\alpha = \alpha'$. At this point cylinder output force reaches the maximum. It is the force at least cylinder has to support under the specified clamping force. The curve P_m is a hyperbola function of α. After the point $\alpha = \alpha'$, P_m increases rapidly and the output force of cylinder P_c decreases. From Fig 5 and reference (Beijing 1979), the following equations are obtained:

$$\alpha' = \frac{1}{\sqrt{3}}\alpha_0 \quad (33)$$

$$\alpha_0 = \sqrt{\frac{6567\times P_{t_{max}}}{L_1(1+\lambda)\times C\times cos\theta}} \quad (34)$$

The clamping system stiffness C is related to the stiffness of the system. It is a combination stiffness of all the toggle components including tie bars, fix platen, moving platen, mold platen, cavity mold, core mold, linkage L_1 and L_2. In this study the clamping system stiffness C is given by:

$$C = \frac{1}{(\frac{1}{ZC_t}+\frac{1}{m_1C_1}+\frac{1}{m_2C_2})\times K} \quad (35)$$

where:
C = total stiffness of clamping system,
λ = relation between linkages $\frac{L_1}{L_2}$,
C_t = tie bar stiffness, equals to $\frac{e_t a_t}{l_t}$,
C_1 & C_2 = linkage stiffness, equals to $\frac{e_1 a_1}{l_1}$ and $\frac{e_2 a_2}{l_2}$ respectively,
Z = number of tie bars (Z =4 for an IMM),
K = compensatory coefficient for clamping system stiffness,
e = young's module, 19300 $\frac{kN}{cm^2}$ is used for tie bars and 17000 $\frac{kN}{cm^2}$ for linkages in this study,
a = cross areas,
l = lengths of tie bars & linkages.

Since the stiffness of fix platen, moving platen, mold platen and mold are too large when comparing to tie bars and linkages. They are ignored in equation (35) and are compensated by a coefficient K. Its value is suggested to be 1.5 by Reference (Beijing 1979).

Following the descriptions of clamping force, the cost function of clamping force maximization can be defined as the equation (36) ~ (38):

$$F = -M \quad (36)$$

Where M is called as clamping force amplitude and is defined to be the ratio of moving plate force P_m and cylinder force P_c, to give:

$$M = \frac{P_m}{P_c} \quad (37)$$

Also from the forces balance on horizontal as shown in Figure 6 (Beijing 1979), M can be derived as function of L_1, L_5, α, γ, θ, ϕ, β as the following equation shown:

$$M = \frac{P_m}{P_c} = \frac{L_5}{L_1} \times \frac{\sin(\alpha+\gamma+\theta+\phi)}{\sin(\alpha+\beta+\theta)} \times \frac{\cos\beta}{\cos\phi} \quad (38)$$

Within these seven variables, for a specified toggle system, the L_1, L_5, γ, θ are unvaried variables. It means these four variables do not change in a mold closed process. While the other three variables α, ϕ, β change their values through the process. In other words, the value M is varying within a mold closed process. A specified α, therefore, is necessary to define first, at that point the force amplitude M of different toggle systems can be compared. The output force of cylinder reaches the maximum at the point $\alpha = \alpha'$ as previous stated. Owing to this reason, $\alpha = \alpha'$ is assigned in equation (38) to calculate M as cost function in the force amplitude optimization case.

Each case has an individual MOST program. All three cases in this study have the same design variables but with their own objective function. In the second case, except these twenty constraints in the first case, owing to $\alpha = \alpha'$ being assigned in equations, two additional constraints shown in the following are added to avoid domain errors:

$$g[21] \equiv \left|\frac{Ed - L_5\sin(\alpha'+\theta+\gamma)}{L_4}\right| - 1 < 0 \quad (39)$$

$$g[22] \equiv \left|\frac{L_1\sin(\alpha'+\theta)-(L_1+L_2)\sin\theta}{L_2}\right| - 1 < 0 \quad (40)$$

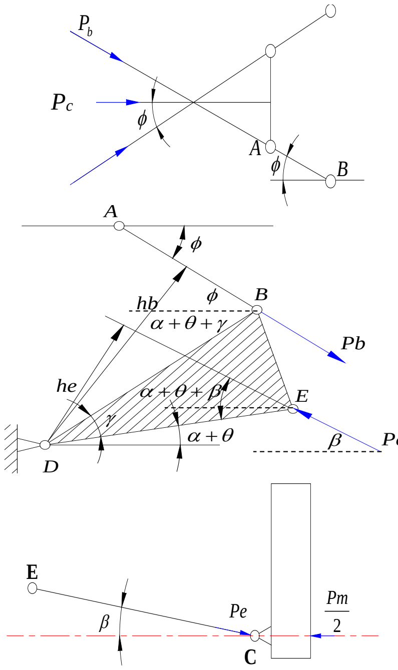

Fig. 6. Free body diagram of linkage and moving platen

Third optimization: optimization of dry cycle velocity

The velocity for the moving platen is obtained by V_m, using the relation of the following equation:

$$\frac{V_m}{V_c} = \frac{dS_m}{dS_c} \quad (41)$$

and according to the virtual displacement method :

$$F_c \cdot dS_c - F_m \cdot dS_m = 0 \quad (42)$$

where also:

$$\frac{V_m}{V_c} = \frac{F_c}{F_m} = \frac{1}{M} \quad (43)$$

Finally, that gives the result of $V_m(\alpha)$ in α conditions:

$$V_m(\alpha) = \frac{L_1 \cdot \sin(\alpha+\beta+\theta) \cdot \cos\phi}{L_5 \cdot \sin(\alpha+\gamma+\theta+\phi) \cdot \cos\beta} \times V_c \quad (44)$$

where V_c is the cylinder input velocity.

From this equation, it is obviously understood that V_m is not constant within a mold closing process even V_c is constant.

An ideal velocity for moving platen is asked to have a slow-fast-slow profile. It starts from a slow speed, then proceed at high speed, and quickly significantly reduced to a very slow speed when close to the mold closing position. The lower velocity in the first stage of mold closing process can prevent vibration and the third stage can protect the mold from collision. According to this request, a desired velocity profile for moving platen is pre-defined. A total square error function Q(x) then is used to represents the error between the desired velocity and real velocity as the following equation shown. To minimize this function means to force the moving platen moves following the desired velocity:

$$\mathrm{Q(x)} = \int_0^{\alpha_{max}} [V(\alpha) - F(\alpha)]^2 \, d\alpha \approx \sum_{i=1}^{N} [V(\alpha_i) - F(\alpha_i)]^2 \qquad (45)$$

where $V(\alpha_i)$ represented the real moving plate velocity and $F(\alpha_i)$ represented the desired moving plate velocity as Figure 7 shown. This curve is defined according to some advices from experienced IMM designers. It can be described as:

(1) The moving platen movement has to start in a lower velocity point, with the purpose to reduce the vibration in the beginning of the displacement, considering 40% of the highest velocity.

(2) No matter how low the velocity start the movement, it has to reach the maximum range throughout the time to save the dry cycle time, the maximum peak has to be over 100%.

(3) Reduce in a minimum level the final velocity in the last part of the process, around 10% to the end in the final point that will protect the mold from high impacts.

In the definition of design variables, a difference with the other cases is that the design variable α_{max} is dropped in this case. The comparison of dry cycle velocity curves of the original and optima design is based on the same distance (from $\alpha = \alpha_{max}$ to $\alpha = 0$, see Figure 7). It means the α_{max} in the original and optima design are the same and this design variable, therefore, is no longer needed in this case.

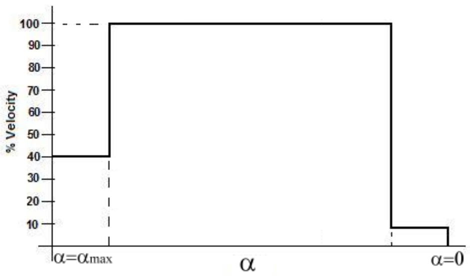

Fig. 7. Desired moving platen velocity used in this study

OPTIMIZATION RESULTS AND DISCUSSION

Relation stroke

The final optima results were tabulated in the Table 2, where it shows the original design and new stroke design dimensions. The cost function of new design is 1.024, indicating it is really better than the original design having cost function 1.125. In order to confirm the feasibility of this optima design, those results were tested with the analysis and simulation on SimWise 4D and Solidworks as shown in Figure 8.

The cylinder stroke (S_c) represents the total distance made by the hydraulic cylinder; the moving platen stroke (S_m) represents the displacement output obtained by the moving platen action (see Fig 2). Comparing the results; there was found that the original design needed 346 mm of cylinder stroke to produce 308 mm moving platen output, and for the new design, 330 mm were needed to produce 322 mm moving platen output. Comparing the objective function of the original and optima designs, almost 10% improvement is obtained.

Table 2. The original and optima design in the stroke optimization case

		Original	Optima
Dimension	L_1(mm)	190	195
	L_2(mm)	242	216
	L_4(mm)	66.5	55
	L_5(mm)	181.47	184.54
	γ(degree)	23	26.8
	θ(degree)	4.6	4
	α_{max}(degree)	110	103.65
Cylinder stroke (mm)		346.84	330
Moving platen stroke (mm)		308.18	322.13
F (Relation stroke)		1.125	1.024

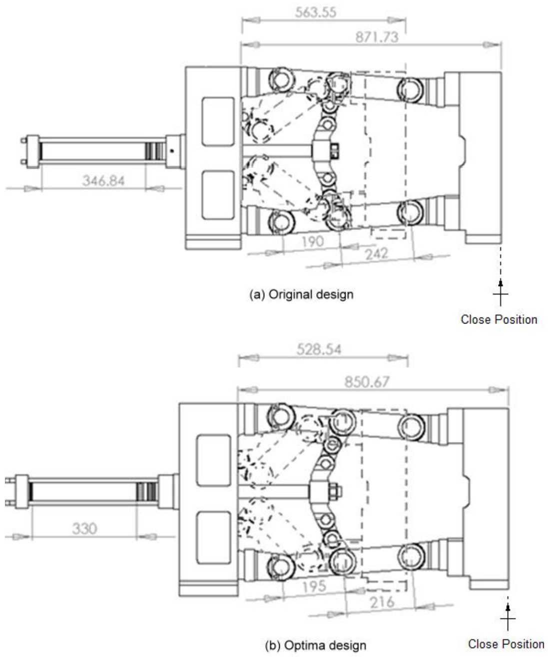

Fig. 8. Confirm the feasibility of this optima result by Solidworks (The broken line represents the open mold position)

Clamping force amplitude

In order to know the improvement of the optima design in clamping force amplitude, the value of clamping force amplitude of both designs are needed to be evaluated. After the optima design is obtained, these data can be used to produce the force amplitude curve as shown in Fig 5. Table 3 shows all the original design values, not only the original linkages dimensions but also the C_p, L_p, C_1, C_2, P_{cm} and the cross areas of the linkages and tie bars. Also, Table 3 shows the result values of C, α_0 and α'. Where $\alpha' = 2.21°$ and $\alpha_0 = 3.83°$. Both angles were important to make all the calculations and α' is the angle where the cylinder needs to output the maximum force.

Table 3. Original clamping force design parameters

E casting	17000000	N/cm^2	L_1	19	cm
E stainless	19300000	N/cm^2	L_2	24.2	cm
L_t	105.6	cm	L_4	6.65	cm
K	1.45		L_5	18.15	cm
C_t	19649081	N/cm	γ	23	deg
C_1	61056842	N/cm	θ	4.6	deg
C_2	68407438	N/cm	eD	14.86	cm
P_{tmax}	1195	kN	dB	5.2	cm
tie bar	107.51	cm^2	dE	8.2	cm
A_1	68.24	cm^2	H	23.6	cm
A_2	97.38	cm^2	P_{cm}	1195	kN
$C =$	15774466	N/cm			
			$1/(4 \times C_p)$	1.27E-08	
α_0	3.82982	deg	$1/C_1$	1.64E-08	
α'	2.21364	deg	$1/C_2$	1.46E-08	

Considering the original design values as starting point, MOST run to find the optima dimensions, as the result shown in Table 4. With these new parameter, the new C_p, L_p, C_1, C_2, C, α_0 and α' as shown in Table 5 can be obtained. Appling these parameters from Table 3 and 5, two clamping force amplitude curves representing the original and optima design can be obtained in Figure 9.

Table 4. The original and optima design in the clamping force amplitude optimization case

		Original	Optima
Dimension	L_1(mm)	190	130
	L_2(mm)	242	190
	L_4(mm)	66.5	75.5
	L_5(mm)	181.47	165.68
	γ(degree)	23	22.33
	θ(degree)	4.6	4.1
	α_{max}(degree)	110	95

From these two curves, it is clear that the clamping force amplitude has been improved in the optima design. If comparing the clamping force amplitude at the position $\alpha' = 2.21°$ for both cases, the value is 28.91 for the original design and is 48.57 for optima design.

Dry cycle velocity

The final optima results were obtained and tabulated in Table 6. Figure 10 shows the mold closing velocity curves of the original and optima design. These two curves are obtained from SimWise 4D. This software can simulate the mold closing motion and output the velocity curves according the CAD models.

Table 5. Optima clamping force design parameters

E casting	17000000	N/cm^2	L_1	13.04	cm
E stainless	19300000	N/cm^2	L_2	19	cm
L_t	94.46	cm	L_4	7.55	cm
K	1.45		L_5	16.568	cm
C_t	21966366	N/cm	γ	22.33	degree
C_1	89004143	N/cm	θ	4.1	degree
C_2	87129473	N/cm	eD	14.86	cm
P_{tmax}	1195	kN	dB	5.2	cm
tie bar	107.51	cm^2	dE	8.2	cm
A_1	68.24	cm^2	H	23.6	cm
A_2	97.38	cm^2	P_{cm}	1195	kN
$C =$	20228264	N/cm			
			$1/(4 \times C_p)$	1.14E-08	
α_0	4.75797	degree	$1/C_1$	1.12E-08	
α'	2.75011	degree	$1/C_2$	1.15E-08	

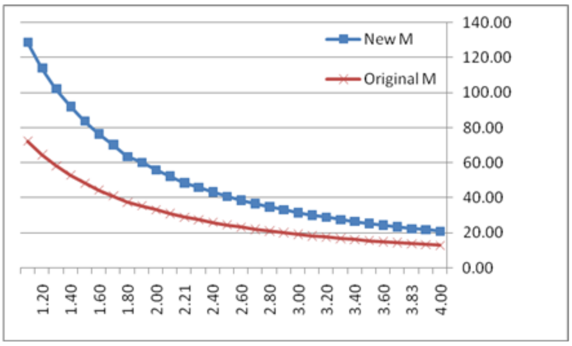

Fig. 9. Clamping Force Amplitude M (Original and New)

Table 6. The original and optima design in the dry cycle velocity optimization case

		Original	Optima
Dimension	L_1(mm)	190	195
	L_2(mm)	242	216.66
	L_4(mm)	66.5	71.93
	L_5(mm)	181.47	195.78
	γ(degree)	23	19.74
	θ(degree)	4.6	4.1

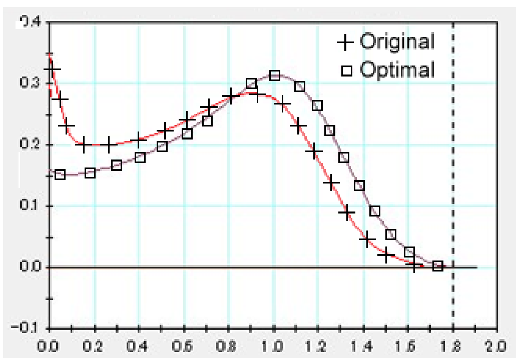

Fig. 10. Dry cycle curves comparison

From Fig 10, it is obvious that the optima curve is closer to the desired curve than the original curve. The most apparent improvement is that in the original design, moving platen starts at 350 mm/s in the starting point while the optima design starts at 165 mm/s. Furthermore, in the middle of the journey, the curve of the original design goes at its maximum velocity 275 mm/s but the optima design reaches 315 mm/s. In other words, the optima design reduces the starting velocity in the initial point, and reaches a higher velocity in the middle stage and is more conformable to the slow-fast-slow profile in velocity request.

CONCLUSIONS

In this paper the clamping mechanism of an injection molding machine is studied by using a six-links mechanism. Three cost functions are given in terms of the relation stroke, clamping force amplitude, and close mold speed of the mechanism. The necessary constraint equations of the toggle clamping system are also presented. From our numerical results the optimal design are really better than the original design: (1) The relation stroke has been increased. (2) The clamping force amplitude has been improved. (3) The starting velocity of moving platen in the initial point is reduced, and reaches a higher velocity in the middle stage.

FUTURE WORKS

Three separate optimization study had been achieved in this study. In fact, we cannot have a design that is optima in all of three requirements. However, the IMM makers still expect to have a general purpose machine whose performance is better than average in these three requirements. A further study therefore will focus on developing an integration optimization system. Within this system, a graphic user interface (GUI) is included and a commercial optimization software is embedded by its API function. These three objective functions are combined to be an integrated objective function along with weighting values. Users can adjust the weighting values flexibly according their requirements to find a suitable design.

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射出成型機肘節夾模機構最佳化設計

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摘 要

本研究的主要目的乃在於進行塑膠射出成型機肘節夾模機構的最佳化設計。利用最佳化方法和三維設計及模擬軟體，針對移動模板行程、夾模力和關模速度三個目標函數進行最佳化設計。從研究結果的數值可以看出，最佳化設計結果確實優於原始設計，包括：(1) 相對於射出油壓缸行程，移動模板行程確實增加。(2) 提高了夾模力放大倍數。(3) 移動模板在初始以一個低的起始速度出發，然後在中間階段以高的速度行進，最後再以極低的速度進行合模。