LAYOUT DESIGN OF A PLATENLESS MOLDING MACHINE

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Abstract

A layout design of a platenless injection molding machine is developed. The machine design is motivated by economics, energy efficiency, compactness, ease of use, and environmental friendliness. The elimination of traditional platens allows for significant performance improvements as well as flexibility of new injection system and mold designs. This paper establishes theoretical feasibility, but also indicates that the design is most appropriate for clamp tonnages less than 150 tons due to actuator power and mold deflection limitations.

Introduction

Traditionally, injection molding machines are composed of three parts: (1) the injection unit, which feeds plastic melt to a (2) mold, which is held closed by (3) a clamping unit. The load paths in the mold and clamping unit of a modern two-platen design are shown in Figure 1. The pressure inside the mold cavity generates a mold opening force which is transferred through shear and compressive stresses to the center of the machine platens. The platens then transfer the load through shear stresses to the tie bar retainers. The retainers, in turn, transfer the load through shear stresses to the tie bars, which should be in pure tension. It should be noted that the mold's guide bushings and the machine tie bars are also subjected to possible lateral loadings. This machine design has a theoretical minimum number of 20 structural components with 8 axial or compressive stresses and 22 shear stresses.

The traditional clamp design is costly to produce and assemble, with the finished machine consuming excessive space and energy. In the present competitive scenario, the industry is looking towards more energy efficient and economical machines. This paper proposes a concept design that reduces the cost of the machine substantially and also makes the mold easier to access. The size of the machine is reduced by eliminating the need for the platens as the traditional method of clamping, and thus would reduce the machine size by more than 50%. By reducing the machine size and weight, the cost of the machine is substantially reduced due to lower material utilization and reduced actuator requirements.

Instead of the traditional design, the mold is supported on one optional stationary platen with internal tension rods connecting the top and bottom mold clamp plates. These tension rods are designed to replace the guide bushings and tie bars, so must be designed to withstand moderate lateral forces as well as very high tensile forces. Furthermore, the mold must be designed to accommodate the tensions rods and resist deflection under maximum melt pressures. This paper will next analyze these design constraints, and then discuss the resulting layout design.

Analysis

The load paths in the platenless machine are shown in Figure 2. The cavity pressure inside the mold cavity generates a mold opening force which is transferred through the mold and retainers by shear stresses to the tension rods. As the name implies, the tension rods are primarily in tension though they are designed to provide lateral alignment of the mold as well. This machine design has a theoretical minimum number of 14 structural components with 4 tensile stresses and 14 shear stresses. Each of the major components will next be considered.

Tension Rods

The tensile stress and lateral deflection of the tension rods were analyzed as [1-3]

$$\sigma = \frac{F}{A_r} \quad (\mathrm{N/mm^2}) \qquad (1)$$

$$A_r = \frac{\pi {D_r}^2}{4} (\mathrm{mm^2}) \qquad (2)$$

$$y_{max} = \frac{W{L_r}^4}{384EI} \quad (\mathrm{mm}) \qquad (3)$$

$$I = \frac{\pi D^4}{64} \quad (\mathrm{mm^4}) \qquad (4)$$

with stress being identified as the dominating constraint. A review of publicly available machines indicates that tie bars are generally designed to an allowable working stress of approximately 40,000 psi. It is noted that this stress level is much lower than the endurance limit for most steels. If 40,000 psi is maintained as an upper stress limit, then the required diameter of the four tension rods as a function of the clamp tonnage is specified in Figure 3. A review of supplier catalogs indicates that standard mold bases, shoulder bushings, and leader pins can be used for these functions up to clamping forces of 150 tons, as is indicated by the green shaded area in Figure 3. Above this load limit, custom components are required that could dissuade mold building for a platenless machine.

End Caps

The end caps are intended to secure the tension rod to the moving side of the mold. As such, the end cap dimensions must transfer the full clamping load from the tension rods to the mold clamping plates. The height of the end cap is calculated by considering the radial contact surface of the tension rod and the internal threads of the end cap (ref. Eq. 5), though ACME thread standards must also be considered. The outer radius r_n and optional flange design of the end cap is calculated considering the contact force between the mold and the end cap and the desired distribution of stress across the mold (ref. Eq. 6).

$$\sigma_y = \frac{F}{2rh\pi} \text{ (N/mm}^2\text{)} \tag{5}$$

$$\sigma_y = \frac{F}{2\pi\left(r_n^2 - r_r^2\right)} \text{ (N/mm}^2\text{)} \tag{6}$$

It may be desirable to use a larger, flanged end cap to distribute the clamping forces towards the center of the mold. The realistic upper limit for the size of the end cap design is not driven by economics as much as the weight requirements to allow ease of operator assembly. Analysis indicates that end caps are not a dominating constraint. As such, the installation of a mold on a platenless machine should require only the placement of the mold and tightening of these four end caps.

Mold Plates

It is observed that by incorporating the tension rods into the mold, several large structures in the machine design and redundant structural alignments are eliminated. However, the mold must be able to accommodate the tension rods and provide minimal deflection under the load of the maximum cavity pressures. While the span between the tension rods in the mold is much less than the span between tie bars in a molding machine, most molds do not have the structural rigidity of machine platens. A slight increase in plate thickness may be allowable, but most practitioners would not tolerate mold plates the thickness of machine platens. As such, economics will dictate the allowable plate thickness.

The mold design must consider the resulting stresses (ref. Eq. 7) as well as deflection (ref. Eq. 8) due to plate bending [4]. Analysis indicates that maintaining a minimum deflection under loads is the dominating constraint.

$$\sigma_p = \frac{0.75W}{t^2\left(\frac{L_s}{w_s} + 1.61\frac{w_s^2}{L_s^2}\right)} \text{ (N/mm}^2\text{)} \tag{7}$$

$$\delta = \frac{0.1422W}{Et^3\left(\frac{L_s}{w_s^3} + \frac{2.21}{L_s^2}\right)} \text{ (mm)} \tag{8}$$

Consider the design of a mold for use with the platenless machine and an average cavity pressure of 100 Mpa (14,500 psi or 7 tons per square inch). As the cavity size increases, both the required clamping force and span increase. However, stiffness increases with the third order of the plate thickness. As such, the required plate thickness will increase to the 3/2 power with the cavity size. Figure 4 shows the required plate thickness for a mold cavity at the center of the mold with a 2:1 aspect ratio and maximum deflection of 0.01 mm (quite stringent). A review of many molds and standard mold bases indicates that 7.5 cm (3 in) mold plate thickness per mold side might be a reasonable upper limit. This area is indicated by the green shaded area of Figure 4 and corresponds to a clamp force of approximately 160 tons. It should be noted that this analysis assumed one large cavity in the center of the mold, and lower deflection or higher attainable pressure is possible if multiple small cavities are spaced between the tension rods.

Clamp Actuators

A ball screw has been initially selected for this layout design due to its relatively high load carrying capability, tight tolerances, and operating efficiency. The ball screw must operate at two different situations for moving the mold at high speed (100 cm/s) with low torque and for clamping the mold at low speed (1 cm/s) with high torque. Depending on the high torque situation and the dynamic load capacity (*C*) of the ball screw and converting the linear speed (n_m) to the rotational speed (n) of the ball screw depending on its pitch the average life of the ball screw is calculated by equations 9 to 11. [5]

$$\textit{Life in Hrs} = L = \frac{C}{F} x10^6 \tag{9}$$

$$\textit{Life in inches} = L_n = \frac{L}{n_m x60} \tag{10}$$

$$\textit{Life in clamping cycles} = L_c = \frac{L_n}{\delta_{clamping}} \tag{11}$$

It was observed that the dynamic load capacity of a ball screw the same diameter of the tension rod was very low. To avoid wear, a much higher diameter ball screw was designed than the minimum calculated. Due to the cost and size of the ball screw, ball nuts, and associated gearing, these components have been placed in and behind the stationary platen.

The operating torque and power of ball screw are calculated by equations 12 and 13. [5]

$$T_d = \frac{FxP}{2\pi e} \text{ (Nm)} \tag{12}$$

$$P_d = \frac{FxP}{2\pi e} x \frac{n}{6.302x10^4} \text{ (Nm)} \tag{13}$$

All four ball screws must be timed to maintain alignment of the mold halves and prevent flash and wear. To achieve this, one ring gear drives all four ball screws. This ring gear is driven by a hydraulic motor. The hydraulic motor must be selected to provide the two different conditions of high speed low torque and low speed high torque. To obtain this without changing gears, a hydraulic motor is selected to satisfy both maximum flow rate and torque requirements. The diameter of the gears is chosen such that even at low motor torque, high ball screw torque can be obtained and at maximum motor power the highest speed can be obtained. These diameters were calculated using the equations 14 and 15.

$$\frac{T_{screw}}{T_{motor}} = \frac{\phi_{screw}}{\phi_{motor}} \qquad (14)$$

$$\frac{\phi_{screw}}{\phi_{motor}} = \frac{N_{motor}}{N_{screw}} \qquad (15)$$

A variable displacement hydraulic pump and accumulator is specified to reduce average energy consumption. The accumulator requirements are performed given the motor displacement and number of revolutions the motor takes to generate the opening stroke of the mold. The number of revolutions of motor is calculated by equations 16 and 17.

$$Revolutions_{Ball\ screw} = Opening\ stroke\ x\ Pitch \qquad (16)$$

$$Revolutions_{motor} = Revolutions_{Ball\ screw}\ x\ \frac{\phi_{motor}}{\phi_{screw}} \qquad (17)$$

Design and Discussion

The foregoing analysis has been conducted for varying machine specifications, with the resulting layout design of a machine with 40 ton clamping force provided in Figures 5 and 6. The authors do not intend to suggest that this machine design is optimal. Rather, this specific layout design is intended to suggest the advantages and opportunities that can be provided by a platenless machine design. From Figures 5 and 6, it is observed that a vertical layout has been selected. This decision was made to allow the direct, vertical installation of molds since no moving platen is required; bending loads on the tension rods by the mold are also avoided. Part removal is intended to be accomplished through automation.

Standard components have been selected for each of the analyzed sub-systems as previously discussed. The specifications of these machine elements are provided in Tables 1 to 4. It is the authors' belief that the machine design could be detailed and built for far less than the cost of a standard machine. The single most massive component for this machine would be the single stationary platen, the mass of which is approximately 200 kg as designed. A hydraulic power system was selected due to its high energy density compared to electric or pneumatic, which is needed to directly and rapidly actuate the mold as previously discussed.

Table 5 shows a comparison between the proposed machine design and two different commercially available machines, a vertical Minijector machine [6] and a horizontal Sumitomo machine [7]. Consuming just 1800 cm^2 (2 sq. ft.) of floor space, the platenless design also provides the highest clamping force. One measure of molding productivity is clamp tonnage capacity per square foot of floor space consumed by the machine. The platenless machine design can provide 10 times the clamping density beyond the best commercially available tiebarless machines, and approximately 15 times the clamping density of more conventional three platen machine designs.

Once the moving platen and the entire back half of the machine are eliminated, entirely new machine layouts become possible. For instance, Figure 7 provides a concept design of a hybrid extrusion-injection machine line. In this case, a single extruder can compound and deliver consistent polymer melt to a series of platenless molds. Each mold can be operated independently, molding different size and geometry parts at varying cycle times. To compensate for the varying injection and pack pressure requirements of the different molds, a melt control technology such as Dynamic Feed™ can be utilized to independently meter flow and pressure to each cavity [8]. The extruder can be readily designed to act as an accumulator and provide a melt cushion, such that short term variations in flow rates to the independent molds do not require any change in the extruder speed.

The design of Figure 7 introduces two major issues, however. Clearly, all four molds operating with a single extruder would necessarily utilize the same material. Thus, there is a significant reduction in flexibility compared to four individual but larger machines that could independently run different materials. Furthermore, the platenless machine also requires that the tension rods in the mold match the ball screws in the machine. As such, it must be known beforehand that a platenless machine will be used, and the exact mold correctly specified. Again, this mold size requirements significantly reduces the flexibility compared to conventional machines, which have a matrix of bolt holes to allow for different sized molds (and often still aren't flexible enough!).

Each molder must ultimately determine the technological strategy by which to compete. The flexibility offered by a set of complementary, conventional molding machines to run different sized molds and different materials is vital to the survival of many custom molders. However, the analysis and resulting platenless machine design described in this paper indicates that conventional machines do not operate anywhere close to the theoretical limits of the process.

Conclusions

A platenless molding machine design was described that uses tension rods to provide the clamping force to the mold. The analysis indicated that standard mold components can be used for molds requiring up to 150 Tons clamping force. The resulting design significantly reduces the size of the machine. For instance, a 40 ton clamping force can be generated by a machine requiring only two square feet of floor space. However, the design does place additional requirements on the mold, and is not able to run molds of varying sizes without adaptation. Nevertheless, such significant gains in performance are possible that the design should be considered in some applications. Currently, the best applications of the platenless machine design may be micro molding, optical molding, injection compression molding, and ultra-large volume production.

References

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7. **http://www.sumitomopm.com/im_series_english.htm** (accessed December 2,2002)
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Table 1: Mold Specifications

Parameters	Mold
Length of Cavity l_c (mm)	76.2
Width of Cavity w_c (mm)	50.8
Thickness of Cavity and Support Plate t (mm)	28.575
Clamp Force F (Tons)	40.862
Tension Rod Diameter D_r (mm)	20
Diameter of End Cap D_n (mm)	30
Height of End Cap h_n (mm)	12
Length of Mold L_m (mm)	254
Width of Mold w_m (mm)	203.2

Table 2: Motor Specifications

Parameters	Values
RPM	2887
Torque (Nm)	121
Max. GPM	30
Length (mm)	130
Diameter (mm)	110

Table 3: Accumulator Specifications

Parameters	Values
Capacity (in^3)	10
Length (mm)	195
Diameter(mm)	57

Table 4: Ball Screw Specifications

Parameters	Values
Diameter (mm)	63.5
Lead (in)	12.7
Rated Dynamic Load Capacity(lbf)	21306
Life (in)	848537.819
Life (clamping cycles)	2.1×10^{11}
Driving Torque for Clamping (N m)	225.031
Power Clamping (HP)	1.492

Table 5: Comparison of Machines

Parameters	MiniJector	Sumitomo (iM33)	Platenless
Tonnage	12.5	33	40
Dimensions W x L x H (in)	48 62.5 64	30 115 65	21 16.2 44
Orientation	Vertical	Horizontal	Vertical
Tons/sq. ft. of Floor Space	0.625	1.377	16.931
Shot Size (oz)	1	1.1	1.1

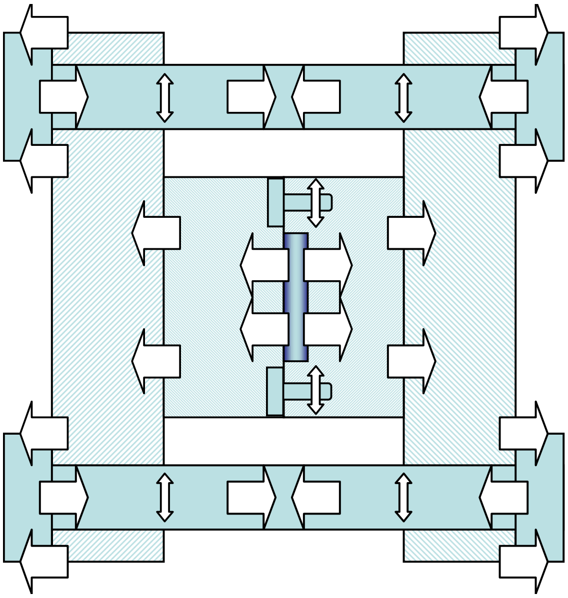

Figure 1: Loading in Conventional Mold, Platen, & Tie Bar

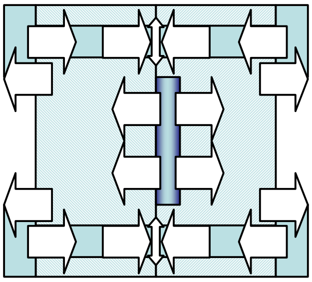

Figure 2: Loading in Proposed Platenless Design

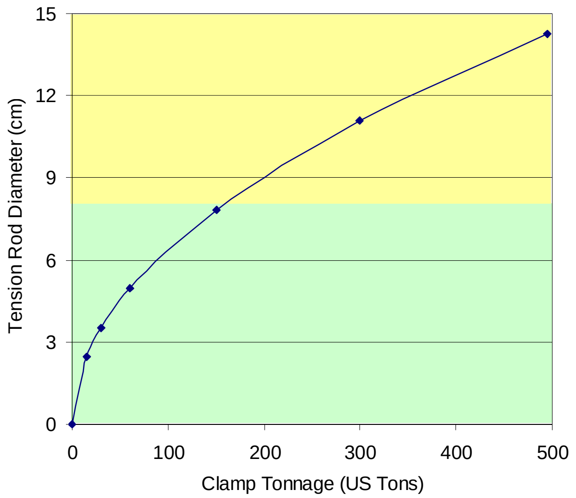

Figure 3: Required Tension Rod Diameter

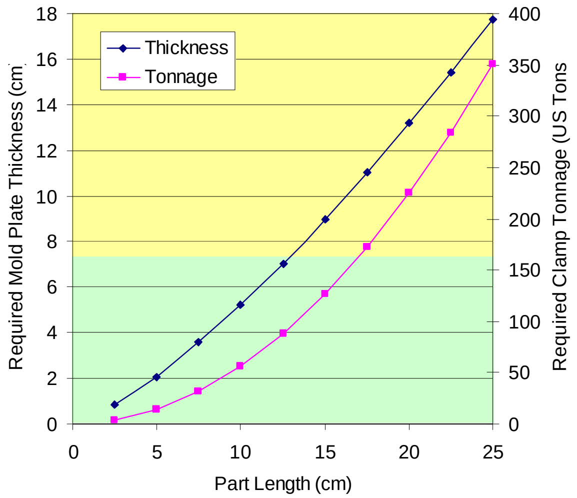

Figure 4: Require Plate Thickness & Clamp Tonnage

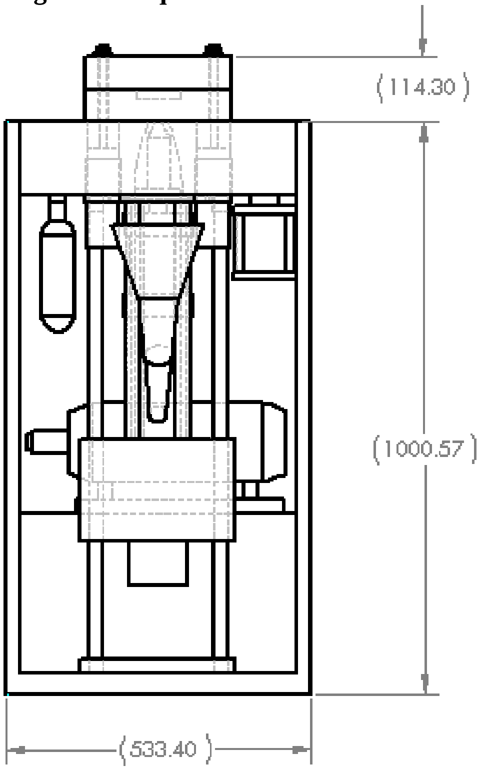

Figure 5: Front View of Platenless Machine

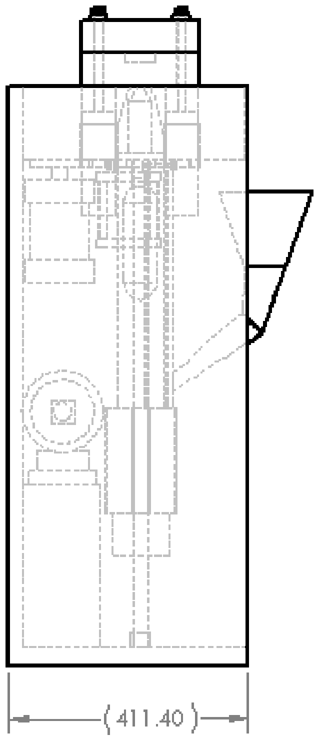

Figure 6: Side Viewof Platenless Machine

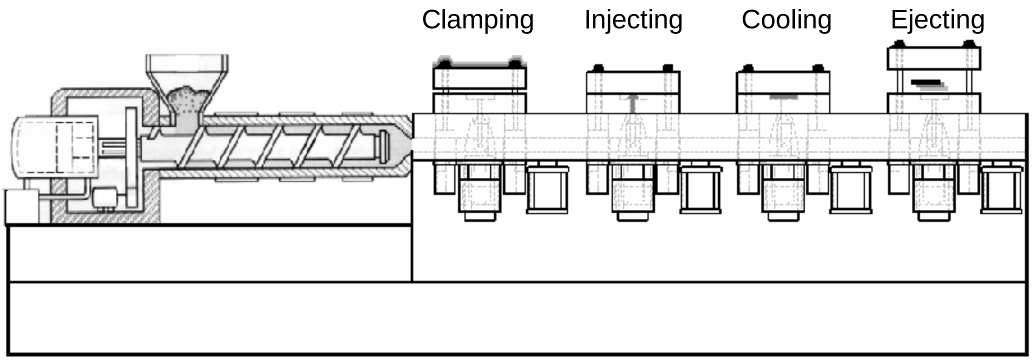

Figure 7: Side View of Platenless Extrusion-Injection Molding Machine Line