THE DESIGN OF A RIG FOR THE DIE-CASTING OF AL-SI PISTON

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ABSTRACT

Pressure die casting is the process where molten metal is forced by pressure into mould. The usual pressure is from 10.3 – 14 MPa. This is the design of an experimental rig for pressure die casting of an Al-Si alloy automobile piston. Two varieties were designed and after applying all necessary design factors (stress analysis) one was optimized and the rig was fabricated and used for casting the piston.

KEYWORDS: Pressure die- casting, Experimental rig, Al-Si piston, design factors.

1. INTRODUCTION

In pressure die casting molten metal is forced by pressure into a metal under a pressure from 0.6-275 Mpa. The casting conforms to the cavity in shape and surface finish. The usual pressure is form 1500 to 2000 psi (10.3 – 14 MPa).

Die casting brass, aluminium and magnetism require higher pressure and melting temperatures and necessitate a change in the melting procedure.

Die casting alloys: A relatively wide range of nonferrous alloys can be die cast. The principal base metals used in order of commercial importance are zinc, aluminium, magnesium, copper, lead and tin. The alloys may be further classified as low temperature alloys and high-temperature alloys. Those having a casting temperature below 813 K (540^o C) such as tin, zinc and lead are in the low-temperature class. The low-temperature alloys have the advantages of lower cost of production and lower maintenance costs. As the casting temperature increases, ferrous alloys dies in the best treated condition required to resist the erosion and heat checking of the die surfaces. The destruction effect of high-temperatures on the dies has been the principal factor in retarding the development of high-temperature die-castings. Other consideration that influence alloy selections are mechanical properties, weight, machinability, resistance to corrosion, surface finish and cost.

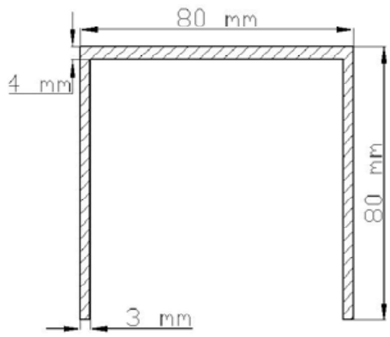
Aluminum base alloys: Many die-castings are made of aluminum alloy because of their lightweight and resistance to corrosion, but they are more difficult to die cast. Because molten alloys of aluminum will attack steel if kept in continuous contact with it, the cold chamber process is generally used. The melting temperature of aluminum alloys is around 813 K (540^o C). The principal elements used as alloys with aluminum are silicon, copper and magnesium. Silicon increases the hardness and corrosion resisting properties; copper improves the mechanical properties slightly and magnesium increases the lightness and resistance to impact. The two principal die casting alloys temperatures are 360^o F and 380^o F (Aluminum Association Designation). Their tensile strength is about 47.000 psi.

2. METHODOLOGY

The design of the rig is based on forcing molten alloy into dies under hydraulic pressure machines operating by this method are two types, one having the plunger in a vertical position and the other in the horizontal position .

3. DESIGN CONSIDERATIONS

Operation cycle is used in a variety of machines that operate at pressure ranging from 5.600 to 22.000 psi (39-150 MPa). These machines are fully hydraulic and semi-automatic. Knowing the geometry of the desired die cast aluminium alloy piston is as follows.



Material to be used for cast piston Al-Si=σ_s=47.000 psi.

Molten-Metal Volume

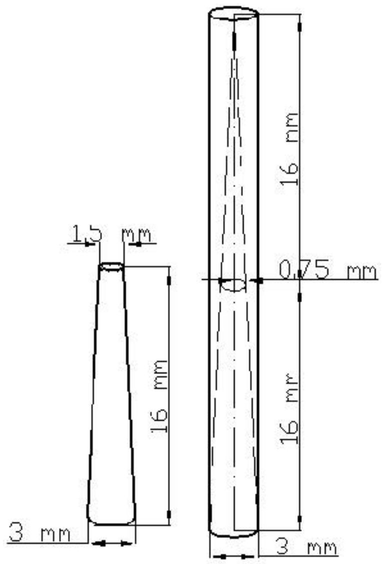
Volume of cast product

Volume=$(\pi/4)(80^2)(80)$- $(\pi/4)(74^2)(76)$= 23956 π mm^3

Total volume of molten metal

V_m=23977π mm^3

<u>Sprue volume</u>



$$\frac{16+h}{1.5}=\frac{h}{0.75}$$

1.5h=16(0.75)+0.75h

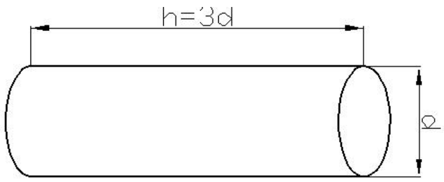
0.75h=16(0.75)=12

h=16

$V_1=1/3\pi r^2(32)-1/3\pi r^2(16)=24\pi-3\pi=21\pi$.

Let the ram's stroke be 3d

Where=$r^2/2$, r^2 => internal diameter of the piston.



$(\pi/4)d^2(3d) = V_m = 23977\pi \Rightarrow d = 31.73787607$

r_2 (internal diameter of cylinder) = d/2 = 15.86893807 mm ≈ 15 will be appropriate

Select d=20 mm from standard.

Design for cylinder

The object of the design for the cylinder is to determine the thickness, which could withstand the working pressure and other design consideration.

Specification:

Assuming that no pressure drop as a result of piston/cylinder friction. This assumption encourages safe design since a pressure drop will tend to reduce the calculated cylinder thickness. The thicker the cylinder however the safe the operation under the required working pressure.

Working pressure of the ram, P_2 = 14 MPa.

At internal radius r_2=d/2, maximum radial struts, $\sigma_x = P_2$ = 14 MPa

The cylinder material is selected: Mild steel.

The maximum tensile stress for a mild-steel will not exceed 86.04 MPa.

Thus, $\sigma_x = P_2$ = 86.04 MPa.

Calculation

The hoop and radial stresses of a cylinder are generally expressed as:

$$\sigma_y = b / r_2^2 + a, \text{ and } \sigma_x = b / r_2^2 - a. \quad (1)$$

Substituting values of σ_x, σ_y and r_2^2 into eq. (1) the constants a, b will be determined.

$$86.04 \cdot 10^6 \cdot \frac{b}{0.015^2} + a \quad (2)$$

$$14 \cdot 10^6 \cdot \frac{b}{0.015^2} - a$$

$$100.04 \cdot 10^6 = \frac{2b}{0.015^2} \rightarrow b = 11254.5 \rightarrow a = 36020000$$

Since the external working pressure of the machine is zero, thus:

$$\sigma_x = b / r_1^2 - a.$$

Therefore, from:

$$0 = b / r_1^2 - a;$$

$$r_1 = (b/a)^{0.5} = 17.676 \text{ mm}$$

Thickness of cylinder: $t = r_1 - r_2$ = 2.767 ≈ 3 mm

Drive Mechanism used on presses:

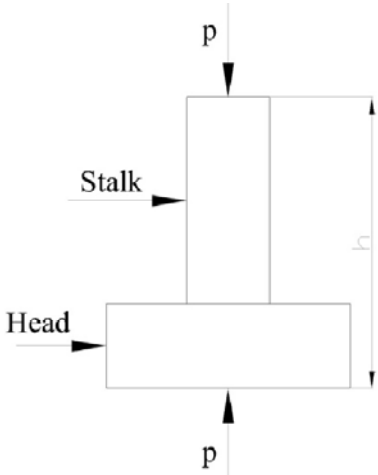
1. Single Crank
2. Eccentric
3. Rack and gear
4. Hydraulic
5. Knuckle joint
6. Toggle drive for blank-holder
7. Screw.

Design for Press Mechanism

Specification

The press mechanism consists of the ram and the hydraulic drive mechanism.

The ram is made up of its stalk and head.



The length of the ram depends on the height which the molten metal occupies in the cylinder which is 3·d in this case, i.e. 3·31.732 ≈ 100 mm.
The ram is taken to be a strut experiencing a longitudinal compression and the crippling load is the working load of the press (14 MPa).
The end condition of the ram is the case III type which case both ends of the ram are guided by the material: MS.

Calculation

In accordance with Eulers theory,

$$\rho = \frac{4 \cdot \pi^2 \cdot E \cdot A \cdot k^2}{l^2}$$

Where E – Young's modulus (N/m^2)
l – the second moment of area Ak^2 (m^4)
L – the length of the column (m)
ρ – the safe load (N)

$$\frac{l}{k} = 2 \cdot \pi \cdot \sqrt{A \cdot \frac{E}{\rho}} = 2 \cdot \pi \cdot \sqrt{\frac{E}{\sigma}}$$

Where σ is the Euler stress, the maximum value of witch must be the stress at the limit of proportionality and l/k=λ is the slenderness ratio.
Since the material for the ram is mild-steel, the Euler stress of mild-steel is 324 MN/m^2 and the Young's modulus E=206 GN/m^2 (GPa).
Therefore,

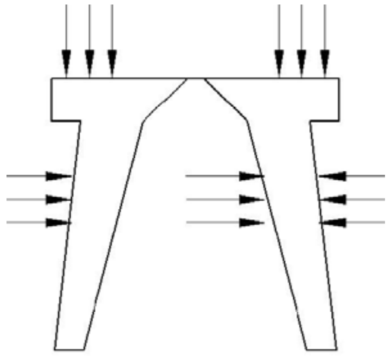
$$\lambda = 2\pi \cdot \sqrt{\frac{E}{\sigma}} = 2\pi \cdot \sqrt{\frac{206 \cdot 10^9}{324 \cdot 10^6}} = 50.43\pi$$

If the stroke length is > 100 mm since the volume height occupied by the molten metal is 100 mm Thus l = 100 mm.
Substituting into equation (2)

$$\frac{l}{k} = 50.43 \;\rightarrow k = 0.00063119\, m$$

k = 0.63 mm.

Design for cavity Insert
Specification:



The cavity insert could be assumed to be a cylinder, subjected to an hoop stress of Ps. Since the metal solidifies under max. 275 MPa, it is taken that $P_2 = 0.6$ MPa.

The cavity insert is made of copper where maximum tensile stress is 200 MPa. The hoop and radial stresses are expressed as:

$$\sigma_y = b / r_2^2 + a, \text{ and } \sigma_x = b / r_2^2 - a.$$

In this case the inner radius of the cavity is the same as the external radius of the cast product.

Thus, $\delta_2 = 80$ mm

Substituting values, b = 641920; a = 99700000

Since $P_1 = 14$ MPa, $\sigma_x = 14$ MPa for external pressure, thus $r_1 = 75.13$ mm; $t = r_2 - r_1 = 5$ mm.

Analysis for bolt design on die casting assembly

Design criteria

The bolt in question is to be subjected to continuous load type of the alternating, fluctuating and repetitive stresses. Thus the bolt material should be a high tensile strength and grade greater than or equal to 8 (i.e. grade > 8, B.S. 3692:1967).

For a uniformly distributed, concentric loading (as in equation) the analysis for bolt design is based on the relation:

$r = W/nA$

Where r – tensile stress in each bolt;

W – load on the component;

A – cross-sectional area of one bolt

n – number of bolts.

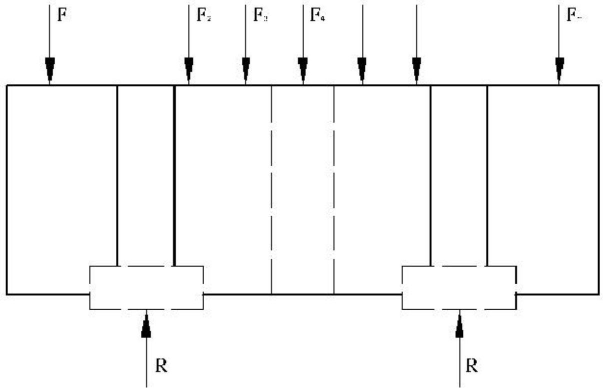
Given: Operating pressure: 5600-22000 psi

Volume of change $2.39975 \cdot \pi \cdot 10^{-2}$ m^3.

Dimensions of each component from thin working drawings, g = 9.81 m/s^2.

Casting operations involves heat transfer, thus the bolt material should be heat resistance alloy steel. From BS specification 3100 302 C35, the material that meet this condition is an alloy with 560 MPa maximum tensile strength.

Loading condition



For equilibrium condition,

$$F_{\sum(1,2,\ldots n)} = F_{\sum(a\ldots z)}$$

Definition , $F_{\sum(1,2,\ldots n)} = F_{change} + F_{operating\,press} + F_1 + F_2 + F_3 + F_4 + F_5 + \ldots + F_{11}$

Where the subscript represents the item numbers on the assembly drawing and F their weights.

Note that $F_3 + F_4 + \ldots + F_{11}$ – static load. Assuming that $F_1 + F_2$ is negligible compared to the force due to operating pressures, then:

$$F_{\sum(1,2,\ldots n)} = F_{change} + F_{operating\,press} + F_S$$

It should be noted that, F_S is dependent on the final dimensions of the items involved and that most of the items are of mild steel material.

Value of F_S

It is important for this analysis to treat the assembly as a "perfect" cylinder having diameter 220 mm (black plate diameter) and of height (333-160) = 173 mm.

Thus F_s is given as:

$$\rho_{ms} = \frac{M_{ms}}{V_{cylinder}}$$

$$V_{cyl} = \frac{\pi \cdot D^2 \cdot H}{4} = \frac{\pi \cdot (0.22)^2 \cdot 0.173}{4} = 0.00657\ m^3$$

Thus,

$M_{ms} = t_{ms} \cdot V_{cyl} = 7850 \cdot 0.00657 = 51.62$ kg;

$F_S = M_{ms} \cdot g = 51.62 \cdot 9.81 = 506.43$ N;

$F_{change} = t_a \cdot$ Vol. of charge $\cdot g = 2700 \cdot 2.4 \cdot \pi \cdot 9.81 = 1996.87$ N.

$F_{op\,pres}$ = Operating Pressure · Cross – sectional area of piston = $386.22 \cdot \frac{\pi(0.06)^2}{4}$=109201.4 N

F_{op2} = Operating pressure x Cross – sectional area of piston = $1517.296 \cdot \pi \cdot (0.06)^2 = 429005.4$ N

Total Downward force on back plate; from eqn (3).

$F_{charge} + F_{ope} + F_{static} = 111704.7$ N or 431508.7 N

These are the total forces acting on the plate at the extreme operating conditions.

Bolt dimension (diameter)

From eqn. (1)

σ = W/nA

nA=W/R

For minimum operation pressures,

nA = 0.000199472

For maximum operating condition

nA = 0.00077051

For the cylindrical section of the bolt, cross sectional area A, is given as:

$$A = \pi r^2$$

Substituting: $nD_{min}^2 = \frac{4(0.000199472)}{\pi} = 0.00025397\, m^2$

$nD_{max}^2 = \frac{4(0.00077051)}{\pi} = 0.000981096\, m^2$.

$$D_{min}^2 = \frac{253.97}{n}$$

$$D_{max}^2 = \frac{984.096}{n}$$

Taking n as 4:

$$D_{min} = \sqrt{\frac{253.97}{4}} = 7.96\, mm$$

$$D_{max} = \sqrt{\frac{981.096}{4}} = 15.66\, mm$$

This shows that the higher the number of bolts, the smaller the diameter (minor) of the bolt and the smaller the number of bolts the higher the diameter of the bolt.

Recommendations

Based on the analysis above, the bolt size should be based on the maximum operating condition, thus, bolt diameter D_{max} is selected for the design of the bolt.

To pick the bolt size, consult B.S. 3643: Part2: 1996 using D_{max} = 15.66 mm as minor diameter.

The designed and fabricated was used for casting the piston varying solidification time and temperature, the end products was now tested for mechanical properties.

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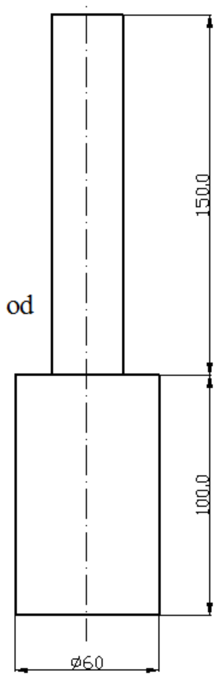
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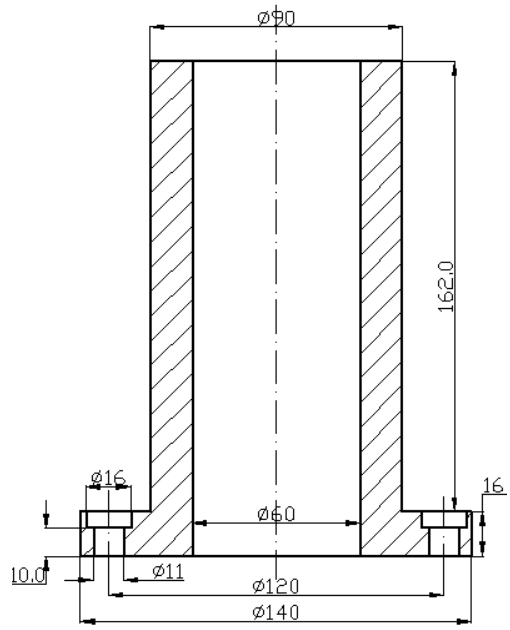
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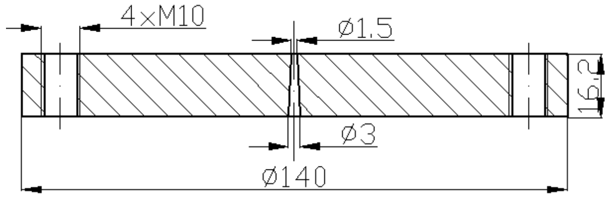
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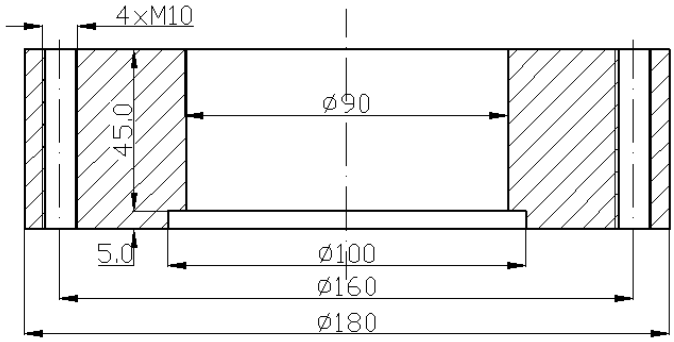
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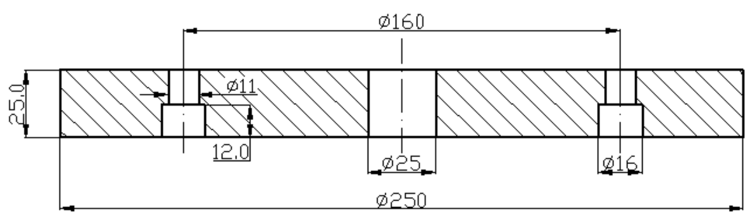
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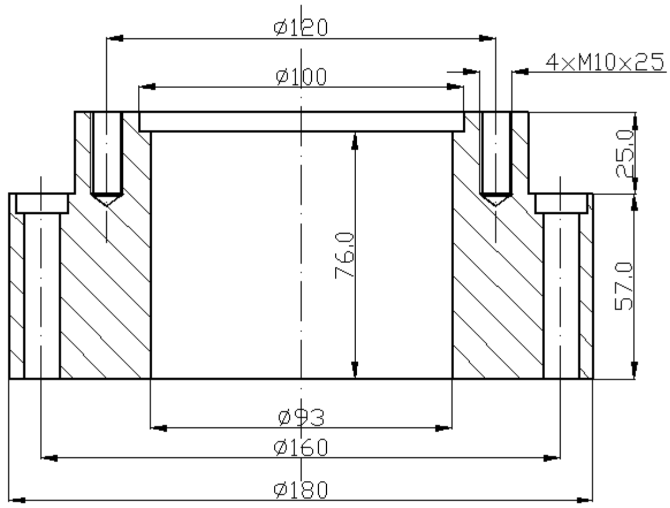
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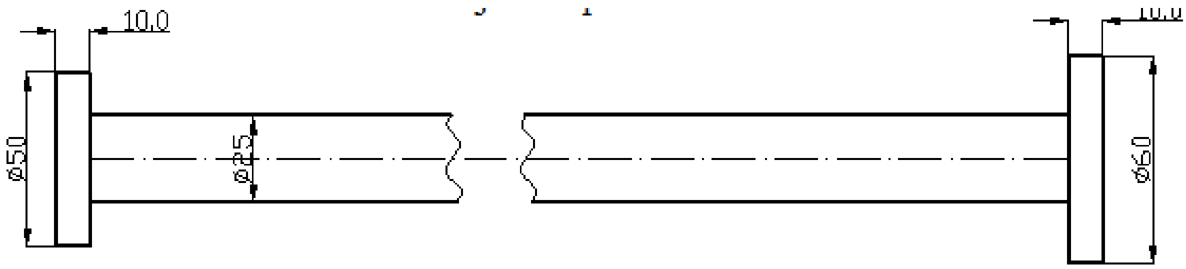
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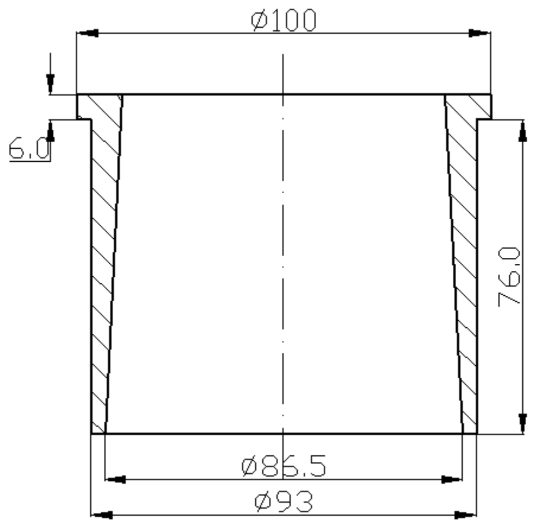
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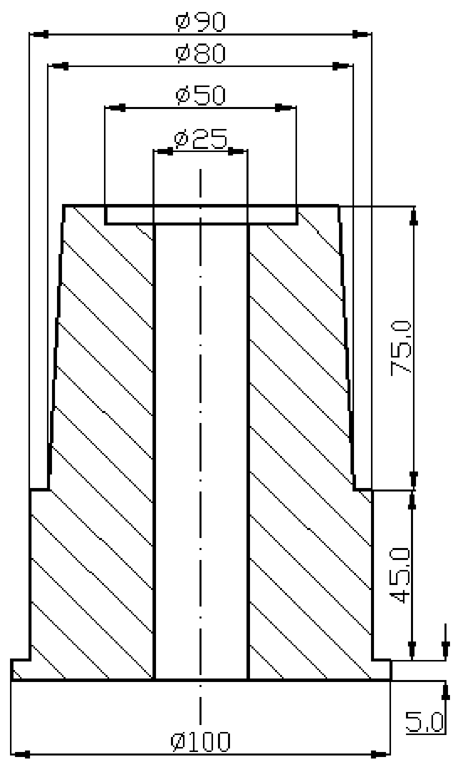
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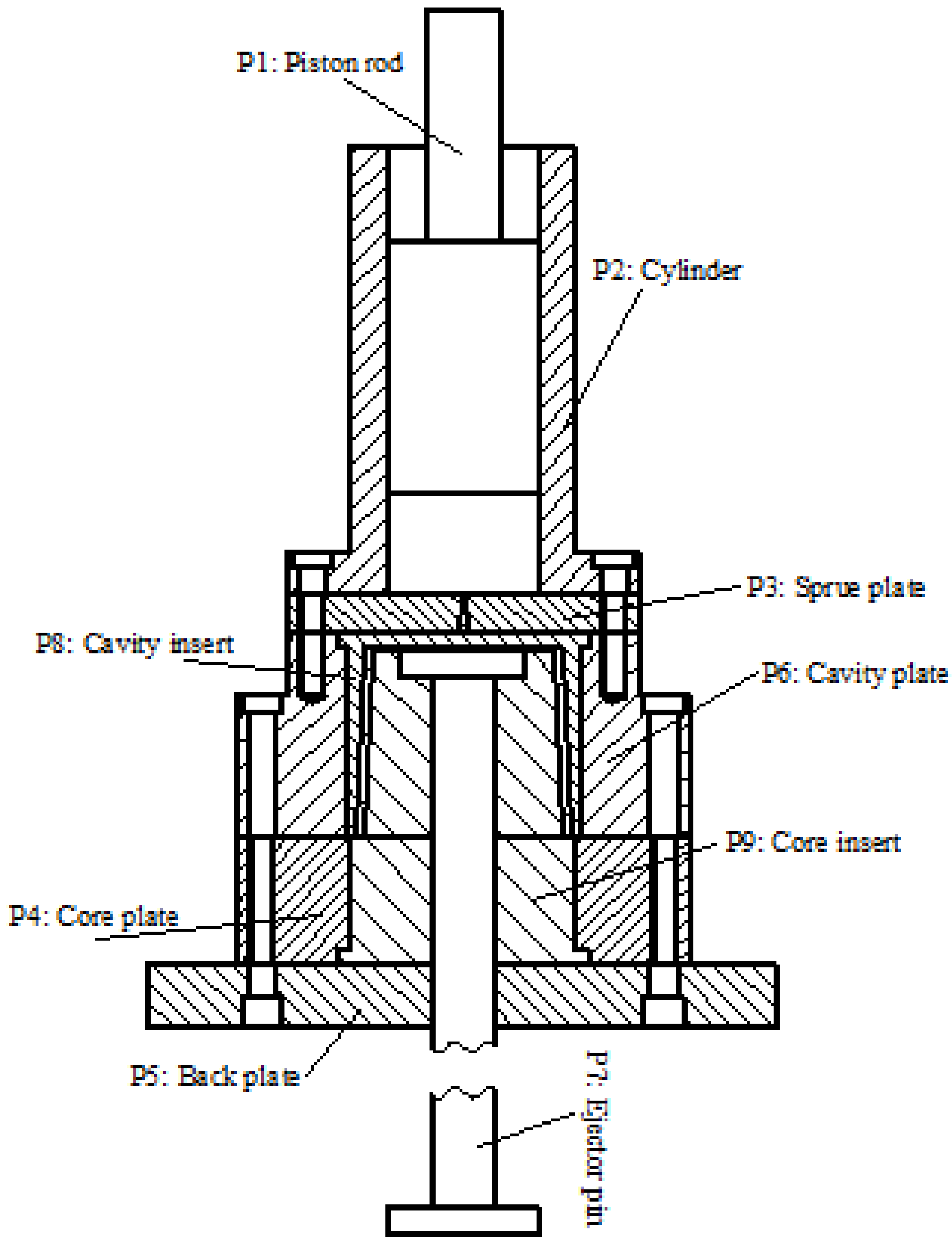
P1: Piston rod
P2: Cylinder
P3: Sprue plate
P8: Cavity insert
P6: Cavity plate
P9: Core insert
P4: Core plate
P5: Back plate
P7: Ejector pin