Modeling the Mechanical Performance of Die Casting Dies

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FINAL REPORT

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Summary

The following report covers work performed at Ohio State on modeling the mechanical performance of dies. The focus of the project was development and particularly verification of finite element techniques used to model and predict displacements and stresses in die casting dies. The work entails a major case study performed with an industrial partner on a production die and laboratory experiments performed at Ohio State. The industrial case study was performed primarily by Ashish Vashist and his masters thesis forms the major portion of the report addressing the case study. Ashish's work was supplemented by additional work on die stresses carried out by Jeeth Kinatingal. The experiments were performed under the supervision of Khalil Kabiri and Adham Ragab with the participation of the entire research group. Adham Ragab prepared the report on the experiments that is included in this document.

The case study die flashed severely after it was moved from its original setup on a 1,000-ton machine, to a new setup on a 2,500-ton cold chamber horizontal die-casting machine. To increase platen coverage on the large machine, and to control die-height, support blocks had been used and ultimately were shown to be a major contributor to the problem. The purpose of OSU study was to conduct simulations on different support setups to understand the sources of flashing and find a design that made operation practical. This provided an excellent opportunity to validate the modeling procedures.

The studies were successful in matching observations with respect to location of flash and qualitatively indicating severity of the problem. Different support arrangements for die setup were simulated and compared with the results observed in the field. These results are discussed in this report.

Experiments carried out at Ohio State University measured the contact pressure between a die and the machine platens in order to confirm the spatial variation in pressure predicted by our modeling. The results obtained are generally quite good, but with a few unexplained anomalies in the measured loads during casting. The details of the experiment are discussed in the report.

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1. Modeling Case Study¹

1.1 Introduction

The past few years have seen a significant increase in the use of numerical techniques in improving the design of die casting dies and die casting machines. There has been an increasing use of numerical techniques such as Finite Element Methods (FEM) and Finite Difference Methods (FDM) in simulating the die casting process. The main reason for their increasing usage is the advent of fast and powerful computers that can handle the numerically time intensive process of solving complex mathematical equations. This in effect has brought numerical techniques to simulate numerous ‘what if’ scenarios rather quickly.

One area for a possible ‘what if’ scenario is the die casting setup on a machine. More specifically, how the die setup can help minimize the distortion at the parting plane due to thermal and mechanical effects, which may cause flashing of molten metal, and help control dimensional accuracy of the part. Previous studies have parametrically analyzed die and machine parameters to minimize such distortions for a standard test part [1]. These test results suggest a good setup for a die on a machine and can help in good design to minimize distortions. However, these results are counter-intuitive and need to be verified in practice. In many cases, such as for the die used in this study, the die and machine are already being used for production and their design cannot be modified due to various constraints. This study provides a good opportunity to test the setup cases which were used by the die caster.

This study focuses on a die casting die currently in use at the die caster’s site. Computational methods were used to analyze combination of machine and die setups to minimize separations at parting plane. The real emphasis was to get some qualitative and quantitative information for a possible improvement or a change in the die setup on the machine.

1.2 Die Casting Process

Die casting process uses a permanent mold for casting metals. A simple open-close die usually consists of two halves made of steel called the cover and ejector die halves. The process is characterized by hot molten metal being forced inside a die cavity by a cylinder and piston arrangement. The piston is usually actuated by a hydraulic mechanism [2]. Mold filling is accomplished at high speeds to ensure rapid filling of the die. Subsequent application of high pressure insures complete fill and helps prevent gas entrapments in the metal. The dies have to withstand the stresses of injection and remove heat from the molten metal to help it solidify. After solidification of the casting, an

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ejection mechanism is used to eject the casting from the die cavity before the next cycle starts again.

Two primary die casting processes used are the hot chamber process and the cold chamber process as shown in Figure 1.1.

The hot chamber process is the original die casting process. The liquid metal is forced into the die cavity through a goose neck. The entire hydraulic plunger actuator arrangement itself sits inside the molten metal to ensure minimum exposure of metal to turbulence and air. It also helps minimize heat loss. The process is commonly used with lower-melting alloys such as that of zinc, lead, tin and magnesium.

A cold chamber machine by contrast has a separate metal holding furnace from where molten metal is poured into the shot sleeve using a ladling system. A plunger then immediately pushes the metal inside the die, completing what is called a shot. As there is very less time of contact between molten metal and shot sleeve, higher temperature alloys such as aluminum and copper are commonly cast using this process. The die under study in this work is mounted on a cold chamber machine for producing aluminum part.

(a)

(b)

Figure 1.1: Schematic drawing showing the difference in shot end machine elements of a) hot chamber and b) cold chamber die casting machine [3]

Figure 1.2: Diagram showing machine elements of a Prince cold chamber die casting machine [4]

For both hot and cold chamber die casting processes, the two die halves mate at what is called a parting surface. This surface could be a simple plane perpendicular to the direction of die closing, or a combination of planes at different angles to clamping direction. The die halves are held closed under adequate clamping force to withstand the high pressure of the incoming metal. This clamping load is provided by the die casting machine, usually in the form of a toggle locking mechanism.

The force of the incoming molten metal places the die cavity under pressure. The clamping force should be more than the pressure on the cavity, multiplied by the projected area of the cavity on a plane perpendicular to the direction of clamp and injection forces. In absence of adequate clamp load, the die is likely to flash, allowing molten metal to shoot out from the sides of the cavity. With the addition of heat from the incoming metal, the die grows. The clamping forces intend to achieve sealing around cavity to avoid flash. However this sealing could be a function of how well the clamp load is transferred through that surface, and also on the extent of distortion of the surface under thermal and mechanical loading conditions.

1.3 The Die Casting Dies and Machine Forces

The die casting process involves mechanical and thermal phenomena that have an effect on the mechanical behavior of the die. Distortion in the dies and deflection of the machine components can have an effect on the quality of the casting produced. The objective is to help minimize these distortions to ensure that the casting obtained is of high quality which is structurally sound and dimensionally accurate.

1.4 Description of the Problem

The die which was analyzed is a single cavity open-close die used to make aluminum channel plates for automatic transmissions in automobiles. The alloy used for casting is commonly used A-380 Aluminum alloy. The parts are characterized by complex geometry and dimensional control of the casting is important. The die halves are 63.5 μm or 0.0025" proud on each side and have cooling lines. The die originally used to run on a 1000-ton horizontal cold chamber die casting machine and was subsequently moved to a new location where it ran on a 2500-ton cold chamber machine. The casting cycle for the new setup was around 60 seconds. The machine and the die shoe are made of 4140 steel. The die inserts are made of H13 tool steel.

The dimensions of the machine platen and the die foot print on the new machine are as follows:

Machine Platen Dimensions

Cover = 107" x 97" x 22"

Ejector = 97" x 97" x 21"

Die Footprint

34" x 32"

In running the small footprint die on a 2500-ton machine, it was found that the die could not be centered on the platens due to the existing shot hole location and the die height, or distance between platens could not be made small enough to accommodate the die.

Because of these constraints, the die was mounted low to align with the shot hole and large spacer blocks were added behind the ejector die box to control height. As the die was not centered on the platen, support structures were designed to increase platen coverage. These supports were two steel blocks with L-shaped cross-section, which were mounted on top of the dies. This arrangement is shown in Figure 1.3. The dimensions of the platen are shown in figure 1.4.

Initial trials with the support block structures resulted in a flash problem. The flash was observed mainly at the top and sides of the die. As the die caster had to run the die on the 2500-ton machine, it was suggested that a numerical simulation be conducted to understand the cause of flashing and to try and solve it.

Figure 1.3: Cover and ejector sides of the machine showing arrangement of ejector and cover dies along with the original support structures – the top L-shaped supports and the spacer block behind the ejector platen

Figure 1.4: A schematic showing the dimensions of ejector and cover platens with original arrangement of the die and supports

1.5 Research objective

The die and machine combination represent a compliant die which is mounted low on a stiff machine. Previous computer simulation studies have suggested that parting separations are minimum when a compliant die is mounted close to the geometric center of a stiff platen [5, 1]. However, these results require experimental verification. The die was already running at the die caster's facility and the study served as a good test of modeling methodology used at Center for Die Casting at the Ohio State University.

Given the constraints in the setup of the die on the machine and the fact that the die was running for production, there was little opportunity for the die caster to analyze the effect of support structures which were added. However, with the use of simulation, various support setups could be analyzed relatively easily by varying parameters such as the clamping tonnage and location and dimensions of supports. The specific objective of the study was to simulate how a small die mounted low on a large machine with high clamping tonnage and additional support structures would behave.

2 LITERATURE REVIEW

Before undertaking any studies in die casting die distortions, it was necessary to review some literature about the die and machine deflections and modeling of the die casting process. A review of some literature relevant to the analysis and modeling of the process is included in this chapter. This chapter includes experimental as well as computational work done on die deflections.

2.1 Characterization of loads on die and machine

The die and the machine together function as a system. Analysis of both is important in understanding the distortions of the die.. The die and the machine are subject to thermal and mechanical loads during the course of a casting cycle. These forces cause the die and machine behavior to change due to deflection and distortion of the components in their assembly.

For studying the loads on the die casting die, the process has to be studied over an entire casting cycle. The forces can broadly be categorized as mechanical loads and thermal loads. [6, 7].

The different stages of a casting cycle are listed below:

1. The die locks up and a clamping force is applied.
2. Hot molten metal is injected at high temperature. The loads involved are due to momentum of incoming metal, heat released during the filling process, and the sudden pressure spike as the cavity becomes completely full.
3. The molten metal is held inside the die and an intensification pressure is applied during solidification.
4. The part is held in the die for a short time to remove additional heat.
5. The die is opened and the part ejected.
6. The die remains open for lubricant and cooling spray and the cycle repeats.

Ahuett-Garza [8] made some recommendations for modeling of deflections taking into account the loads during the cycle. The conclusion was that the momentum from molten metal and hydrodynamic loads can only be important for poorly designed structures or uncharacteristic filling condition. Injection pressures may produce large deflections based on the dynamic response of die and machine. However, in majority of cases, these maximum deflections are achieved at intensification pressure which was considered important in the study of deflections. Also heat released from solidification is important to study thermal growth and its effect on die deflections.

Thermal and mechanical loads over many cycles may or may not effect deflection of dies during normal course of running. Over many cycles, thermal fatigue may crack the die surface, causing checking failure. Coining of dies and catastrophic failure could occur due to constant mechanical fatigue.

The clamping force has to hold the dies shut and withstand the metal pressure. However, simple free body relations balancing forces may not indicate how that clamp force is being transferred across the die face. Experience has also shown that balancing of forces on the tie bars may not guarantee flush parting surface. In final analysis, the clamp should not only hold the dies together, but should also be able to seal the parting surface to prevent shooting of metal across its face.

Another point that has to be accounted for is that the load path across the parting surface will change as the die grows by thermal expansion due to heat which is accumulated over multiple cycles. This is the main mechanism that explains why a cold die behaves differently than one which has been 'run' for many cycles. Typically, die casting process reaches a steady state after tens of warming cycles. Most notably the temperature in the die stabilizes assuming that the temperature of cooling water is stable and repeatable spray cooling can be achieved. Thus a quasi steady state is maintained over a large number of production cycles and this is the stage which can be simulated. The number of cycles to reach of quasi steady state can vary over different dies with varying cooling and spray conditions and even ambient temperatures.

2.2 Thermal Phenomena In Die Casting

Papai and Mobley [9] measured temperature fields for a H13 steel die and computed heat flux values at die and molten metal or die and casting interface. For a cold die, the convective heat transfer coefficient is very high causing die cavity surface to heat up rapidly. This causes the cavity surface to expand or the die grows. These expansions can cause thermal stresses at the surface. The heat transfers across the die and casting interface are affected by the fluid lubrication as well as coatings, if used on the die cavity.

With the growth of the die at the parting surface, in most cases the region around the cavity seals more than the edges. The edges tend to 'flare up' with most of the load transferred at the region around the cavity. This fact has been demonstrated by studies at the Ohio State University [5, 10-15] as well as by studies at General Motors [16]

Another thermal aspect concerns the cooling which is typically achieved by the cooling lines, free convection to air, and forced convection due to coolant spray or air. Location of cooling lines, coolant flow rate and the scaling inside the cooling line controls the effectiveness of the cooling line. The coolant spray cools the die in preparation of the next cycle.

2.3 Experimental Studies in Die Deflection

Prince et al [17] studied the movement of die and its distortion during clamp of die casting die. The study was conducted for a standard die casting design. Movement of parting plane and the slides was measured. The results indicated that increased locking forces can increase the locking performance of the die but only to a certain extent. Excessive locking forces along with effects of heat cause the die to distort. The results suggested lesser distortions for dies with flat parting planes and with lesser locking forces.

Another study by Klein [18] in Germany, published in English as a translation by Takach suggested that loads on four tie bars should be kept equal to fully utilize the clamping forces. The study also suggested that flash could be caused due to uneven tie bar loading. An important factor in considering tie bar loads was that the loading center of die should coincide with that of the platen and that the footprint of the die on the platen should be high. Finally, the study gave some reasons because of which the effectiveness of the clamping force can be reduced. These could be due to the die design, location of die on the platen, stiffness of the machine, uneven preloading on tie bars and thermal effects on the die parting surface. Figure 2.1 shows the forces and moments, as measured on the die casting machine by Klein.

An important consideration in the design of die casting machine is its ability to provide the necessary locking force. The mechanical advantage achieved by the toggle mechanism moves the platen in its locking position while the tie bars stretch to provide the necessary clamping force. These forces can be determined by measuring the stretch of the tie bars.

A good example would be the procedure used by Idra Prince [4] a commercial die casting machine builder. Idra Prince uses tie bar load indicators to measure tie bar stretch. They use a dial indicator and an indicator rod arrangement. The rod is pushed inside a tapered hole along the axis of tie bars, which allows for movement when tie bars stretch. The dial indicator could also be replaced by linear optical encoders, strain gauges or LVDTs. Balancing the loads on tie bars is usually done by placing a master test block which is centered and covers 50% of platen area between tie bars. Tie bars are balanced manually through individual tie bar adjustments, or automatically.

Figure 2.1: Diagram showing direction of forces and moments as measured on the die casting machine by Klein et al [18]

2.4 Die Deflection Studies At The Ohio State University

With funding from the North American Die Casting Association (NADCA) and the Cast Metals Consortium (CMC) along with the Department of Energy (DOE), research projects were undertaken to predict deflection of die casting dies.

Ahuett-Garza [8] did the initial work in characterizing loads in die casting. Padiyar [19] and Hegde [20] addressed the issues to simulate die casting deflections. The modeling approach was developed and tested using ABAQUS [21, 22]. Boundary conditions for the simulation were developed based on experimental data and field observation. These procedures were general enough to be applied to a variety of die and machine combinations. The separation value at the parting surface was established as the indicator of distortion of the die.

The main concern was to find out the interaction of various loads and sort out the ones that are relevant in the deflection of the die. The initial model used rigid supports at back of cover die. The machine supports like platens were later added. Clamping forces and hydrostatic pressures applied on the cavity were used to describe the mechanical forces.

For applying thermal loads, it was assumed that heat flux over the cavity decreases exponentially with time and at ejection has decayed to around 1% of the total heat flux at the beginning of injection [20]. The heat flux values were calculated by dividing the casting and the cavity areas into different regions. The ejection temperature values were checked by using MAGMAsoft [23].

Choudhury [13] used the die casting machine supports to find its effects on overall deflection of the dies. The ejector and cover platens along with the tie bars were added to provide a more accurate model. Spring elements were used to represent the elastic supports behind the platens.

Dedhia et al [24] conducted a parametric study to determine structural variables which effect die deflections. Design of experiments was used to identify these variables. The thickness of platen and cover die, and the location of die on the platen appeared to be the main factors controlling parting surface separations. Kesavan [10] studied the structural behavior of die and the machine. The finite element modeling procedures were refined to use thermal effects over multiple cycles. The boundary conditions for the dies and the platens were also modified.

The effect of structural variables was studied by various authors [14, 15, 11, 1]. An array of variables was used to study die deflections on die and machine components. The variables which were analyzed were die size, die thickness, platen thickness, ratio of thickness of steel behind the insert to total die thickness (thickness ratio), and die location.

2.5 Results From Parametric Analysis Of Die Casting Design Parameters

The previous research suggested when parting surface separation is being considered, it is likely to be minimum with a small footprint and a compliant die which is mounted near the geometric center of a thick and stiff platen. Some of these results were somewhat counterintuitive and required experimental verification.

The parametric study was based on an experimental array analysis of different setups using finite element method. In spite of lack of real life verification, the results did provide a guideline which a die casting die and machine designer can use while designing.

A summary of results of these die casting deflection studies was published by Miller [25]. It emphasized the need to use stiff platens and die shoe coverage of at least 50 percent of the platen area to minimize parting plane separations.

3 CASE STUDY MODEL DESCRIPTION

3.1 *Model Assumptions and Simplifications*

While building the FEM model, some basic assumptions and simplifications were made to facilitate model tractability. The part has a complex geometry which is difficult and time consuming to model as a finite element analysis. However, as a simplification, the part was considered essentially as a flat plate while keeping the volume almost the same.

In previous studies [1] stiffness of the toggle was used. A displacement boundary condition with a spring element was used to apply clamping. As the geometry of the toggle mechanism was not available, the stiffness of the toggle could not be readily calculated. For simplification, the clamping force was modeled as a pressure boundary condition on toggle pad areas behind the ejector platen. This pressure boundary condition is independent of distortion or displacement and does not account for the stiffness of the toggle linkages.

Previous studies by Kesavan [10] and results from the research group have indicated comparable results between models using pressure boundary condition and others using toggle stiffness – displacement model. However, the pressure condition cannot adequately account for load changes due to thermal growth of the die on parting surface.

The third platen, or the adjustable platen behind the ejector platen was not modeled explicitly. Instead, the end nodes of the tie bars were constrained to achieve the effect of the third platen.

All the support blocks are a combination of pieces welded together. However, all welded pieces were modeled as a single piece of support structure. Similarly, the spacer block, and new spacer block with gussets (to be described later in the chapter), are modeled as a single piece. The bolted joints were modeled as either tied nodes, or modeled as a single piece. A summary of actual joints and as modeled joints are shown in Table 3.1. A schematic showing the various boundary conditions used for the analysis is shown in Figure 3.1.

No.	Joint between pieces	Actual joining process	Modeled as/with
1	Cover platen with cover die	Bolted with T-bolts	Tied corner nodes
2	Plate between cover die with cover platen	Bolted	Modeled as one cover die
3	Cover die with cover insert	Bolted	Tied corner nodes
4	Cover insert with cover side shot block	Bolted	Modeled as one cover insert
5	Cover insert with brain insert which sits inside cover insert	Bolted	Modeled as one cover insert
6	Cover side support pillars behind ejector die	Bolted to cover die	Modeled as one cover die
7	Ejector die, ejector mechanism box with support pillars	Bolted together	Modeled as one ejector die
8	Ejector die with ejector insert	Bolted	Tied corner nodes
9	Ejector insert with ejector side shot block	Bolted	Modeled as one cover insert
10	L-shaped support pieces	Welded together	Modeled as one support
11	Spacer block pieces, including gusset support	Welded together	Modeled as one support

Table 3.1: Summary of bolted and welded joints modeled as single pieces or with tied corner nodes

Figure 3.1: Schematic showing the standard finite element technique to model the machine with dies and support structures. The figure also shows the boundary conditions used and the coordinate axis

3.2 Building the Finite Element Model

The geometric of die and machine was modeled using SDRC/I-DEAS [26]. Meshing and boundary condition were also applied in I-DEAS. The model was subsequently exported to ABAQUS [21, 22] as an input file. In case of thermal simulation, time transient heat transfer simulation was solved by defining cycle times and copied over many cycles. Further details of the thermal model are explained in section 3.4.

Since there was no symmetry in the part geometry, the full model was simulated. Most of the die and machine components were modeled as standard solid elements. The tie bar and the leader pins were modeled as beam elements. The tie bars were constrained in space behind the ejector platen, On the cover side, a shell element was used to simulate a nut holding the tie bar. This triangular shell element was tied at the edges with the cover platen. The central node was connected to the tie bar to hold the two pieces together. A thickness equal to the thickness of tie bars was provided to the shell nut to ensure adequate rigidity in the joint. Figure 3.2 illustrates the technique used to simulate nut on the cover platen.

Cavity intensification pressure was simulated by applying constant pressure boundary condition on the cavity surface. As a simplification, the cavity pressure was assumed to be constant throughout the cavity. A pressure value of 11,000 psi (76 MPa) was used. Clamping tonnages of 1400 American ton as well as the full clamping tonnage of 2500 American ton was used in the simulation. A value of 1400 ton was used as it was closer to the original clamping tonnage. Both the clamping tonnage and pressure values were as suggested by the die caster. The model of the dies and the leader pins is shown in Figure 3.3.

The coordinate system used in the model follows the following convention. As was usually the convention with all the previous die deflection studies at Ohio State University, the direction along the tie bars was kept as the Z direction, while the XY plane lies on the parting plane which in this case, is perpendicular to the direction of the tie bars. The coordinate axes are shown in Figure 3.3. The figures in this chapter also indicate the coordinate system. Axes 1, 2 and 3 are for X, Y and Z respectively.

Figure 3.2: Modeling of the tie nut using shell element. Also shown are the platen (solid elements) and tie bar as beam elements

Figure 3.3: Schematic showing the standard finite element technique to model the dies and leader pins. The figure also shows the pressure boundary conditions used for cavity pressure

3.3 Material properties and other parameters used

Only the inserts were modeled as H13 steel. The die shoes, as well as the machine parts were modeled as 4140 steel. A constant Coulomb's friction value of 0.15 was used throughout the analysis. The values of parting plane separations were found to be relatively insensitive to friction values for low friction. The values of material properties used for static mechanic analysis are tabulated in Table 3.2. These are typical values for most steel alloys.

Material Property	H-13 Steel	4140 Steel
Modulus of Elasticity (MPa)	2068	2068
Density (Kg/m ³)	7820	7820

Table 3.2: Material properties used for static mechanical only simulation without heat [2]

3.4 Thermal model

3.4.1 Introduction

As the initial problem with flash was suspected to be a pure mechanical one, as a first approximation, no thermal loads were applied to the model. Thermal loads, including the cooling lines were subsequently added to the model with all the cooling lines. Figure 3.4 shows the location of cooling lines as modeled in the thermal model. The ejector side has 9 cooling lines while the cover side has 7. The cooling lines are explicitly modeled in the analysis.

The alloy used is Aluminum A-380. The standard thermal properties used for the steel and aluminum alloy are shown in Table 3.3 below.

Thermal Material Property	H-13 Steel		4140 Steel		Aluminum 380 Alloy
Thermal expansion (m/mK)	1.17 x 10 ⁻⁵		1.17 x 10 ⁻⁵		2.25 x 10 ⁻⁵
Conductivity (W/m-K)	29		40		109
Latent heat (J/Kg)	N/A		N/A		3.89 x 10 ⁺⁵
Solidus temperature (°C)	N/A		N/A		538
Liquidus temperature (°C)	N/A		N/A		593
Specific Heat (J/Kg/°C)	Specific Heat	Temp (°C)	Specific Heat	Temp (°C)	Specific Heat
	459	23	473	150	963
	519	200	473	200	
	588	400	519	350	
	726	600	519	400	
	905.	700	561	550	
	1151	760	561	600	

Table 3.3: The thermal material properties used for simulation purposes [2]

3.4.2 Using a Heat Transfer Coefficient Boundary Condition

The approach used in this research is to apply thermal loads using a casting and defining a heat transfer coefficient between the die and the casting. By defining this value of heat transfer coefficient, along with latent heat, solidus and liquidus temperature of the alloy, one can accurately simulate the transient transfer of heat from the casting to the die.

However in such a case, finding the correct value of heat transfer coefficient would be of importance. The value of this heat transfer coefficient decreases with a formation of gap between the casting and the inserts and increases with increase in temperature. The study of a gap formation and of temperature dependence on heat transfer coefficient is beyond the scope of this study.

As a simplification, a one dimensional steady state conduction model can be used to analyze for the heat transfer through a die surrounded by molten metal and air. [27]. For steady state heat transfer in one-dimension with no heat generation, the heat flux is constant. This is shown by equation below.

$$\frac{d}{dx} \left(k \frac{dT}{dx} \right) = 0 \quad (3.1)$$

Where

$$k \frac{dT}{dx} = \text{heat flux in one dimension}$$

$$k = \text{thermal conductivity}$$

$$\frac{dT}{dx} = \text{rate of change of temperature along } x$$

A schematic for the one-dimensional heat transfer can be as shown in Figure 3.5.

(a)

(b)

Figure 3.4: A schematic showing the location of cooling lines on (a) Ejector die – 9 lines, and (b) Cover die – 7 lines

Figure 3.5: One dimensional heat transfer from molten metal across the die to air [27]

The thermal resistance model can be built by defining the thermal resistance due to conduction through the die, and convection at the die-metal and die-air interfaces.

Thermal resistance due to conduction can be expressed as

$$R_{t,cond} \equiv \frac{T_{die,metal} - T_{die,air}}{q_x} = \frac{L}{kA} \quad (3.2)$$

Where

q_x = heat transfer rate in x direction

L = thickness of die

$T_{die,metal}, T_{die,air}$ = surface temperatures

k = thermal conductivity of die

A = area across which heat transfer occurs

Similarly thermal resistance for convection can be given by Newton's law.

$$R_{t, conv} \equiv \frac{T_{surface} - T_{fluid}}{q} = \frac{1}{hA} \quad (3.3)$$

Where

h = heat transfer coefficient across the surface

q = heat transfer rate

Total heat transfer rate may also be expressed as

$$q = \frac{T_{metal} - T_{air}}{R_{tot}}$$

R_{tot} the total thermal resistance may be expressed by combining resistances due to conduction and convection in parallel.

$$R_{tot} = \frac{1}{h_1 A} + \frac{L}{kA} + \frac{1}{h_2 A}$$

Where h_1 and h_2 are heat transfer coefficients for die-metal and die-air interfaces respectively.

Assuming that k is constant and value of L is large, the term $\frac{L}{kA}$ would be the dominant factor in the above equation. As value of h_1 increases it becomes less and less significant in the equation.

Different authors have experimentally found different values of this heat transfer coefficient under different conditions. Lin and Righi [28] have reported a heat transfer coefficient of around 1000-10000 W/m²-K for A-356 Aluminum alloy and die interface. MAGMAsoft [23] uses a heat transfer coefficient of between 3000 to 7000 W/m²-K for a die-alloy interface similar to that of A-380 and H13 steel. Another study by Hwang [29] measured a value ranging from 585 to 710 W/m²-K varying with time.

Given such a variable number of values, an average value of 5000 W/m²-K was chosen. Also, as the die is relatively thick, this was a reasonable number. A detailed study of heat transfer across the die-metal was beyond the scope of the study and the temperature field obtained was good for a qualitative comparison of parting plane separations between different support structures and clamping loads. The same temperature field was applied for all such comparisons.

ABAQUS can apply a heat transfer coefficient which varies with gap as well as with temperature. It is called the gap conductance. For the present analysis, as condition of no-gap was assumed, the value of gap conductance was taken as constant

3.4.3 Modeling the Cooling Lines

The cooling lines were modeled using techniques used in earlier studies [5, 6]. The lines were modeled explicitly as a hole thorough the dies and the inserts. A convective heat transfer coefficient of 5000 W/m²-K was applied on the free element surfaces of cooling line elements.

3.4.4 Modeling the Spraying of the Die

The spray was modeled as a negative heat flux boundary condition and applied on the free element surface of the cavity. It was assumed that around 20% of heat added during solidification is removed by spray. A simple calculation was based on the conservation of energy equation.

Given that,

Density of A-380, $\rho = 2760 \text{ Kg/m}^3$

Volume of casting, $V = 1.093 \times 10^{-3} \text{ m}^3$

(The volume is not of actual casting but of the simplified geometric model of the casting, and includes runner, gates and biscuit.)

Specific heat of liquid metal, $C_{p,l} = 560 \text{ J/Kg-K}$

Specific heat of solid metal, $C_{p,s} = 5.2 \text{ J/Kg-K}$

Injection Temperature, $T_{\text{injection}} = 650 \text{ }^{\circ}\text{C}$

(The die caster suggested a value of $680 \text{ }^{\circ}\text{C}$ for the melt temperature. A value of injection temperature of $650 \text{ }^{\circ}\text{C}$ was chosen as it is a typical value for such a die casting processe)

Average Ejection Temperature, $T_{\text{ejection}} = 200 \text{ }^{\circ}\text{C}$

(The value of ejection temperatures was obtained by the averaging the values of temperatures of nodes for the casting for a typical simulation run.)

$T_{\text{liquidus}} = 593 \text{ }^{\circ}\text{C}$

$T_{\text{solidus}} = 538 \text{ }^{\circ}\text{C}$

Latent Heat = $3.89 \times 10^5 \text{ J/Kg}$

Now, total heat added to the die by casting is given by the following equation,

$$\begin{aligned} Q_{\text{total}} &= Q_{T_{\text{injection}} \rightarrow T_{\text{liquidus}}} + Q_{\text{latent}} + Q_{T_{\text{solidus}} \rightarrow T_{\text{ejection}}} \\ &= m_{\text{casting}} c_{p,l} (T_{\text{injection}} - T_{\text{liquidus}}) + m_{\text{casting}} L + m_{\text{casting}} c_{p,s} (T_{\text{solidus}} - T_{\text{ejection}}) \end{aligned} \quad (3.4)$$

Where,

$m_{\text{casting}} = \text{mass of casting} (= \rho \times V)$

Putting the values in Equation 3.2 above, total heat released by the casting,

$$Q_{\text{total}} = 2.2 \times 10^6 \text{ J}$$

Total area exposed to casting, $A = 2.72 \times 10^{-1} \text{ m}^2$

Duration of spary, $t = 11 \text{ s}$

Coolant or spray factor, $c = 0.2$, or 20%

(The spray factor is the ratio of assumed heat removed by spray to the value of heat added by the hot metal)

Total negative heat flux removed by spray in each cycle,

$$q_{\text{spary}} = \frac{Q_{\text{total}} \times c}{t} = 1.47 \times 10^5 \text{ W/Kg-K}$$

3.4.5 Solid Elements Used For Thermal Simulations

All the mechanical simulations without heat were conducted by using modified 10-noded tetrahedron solid element and suitable beam and shell elements. However, limitations were discovered in using the 10-noded elements for running transient heat transfer simulation. Spurious and incorrect temperature values were found in the output. On further investigation the ABAQUS manual mentions these as spurious “non-physical” oscillations due to small time increments. The ABAQUS manual [21, 22] provides a simple relation between the minimum usable time increment and the element size for a transient heat transfer analysis with second-order elements such as that of 10-noded elements.

$$\Delta t > \frac{\rho c}{6k} \Delta l^2 \quad (3.5)$$

Where,

Δt = time increment

ρ = density

k = thermal conductivity

Δl = typical element dimension

The second-order element mesh which was used failed the above test. First order elements like 4-noded elements eliminate or reduced such spurious outputs to a minimum. The problem was further reduced by using a fine mesh as recommended by ABAQUS. However, these first-order elements were not found suitable for defining contact conditions in a mechanical simulation.

8-noded brick elements were found to be very stable in transient heat transfer simulation as well as in convergence with contact conditions under mechanical loads. However, building brick elements is a time intensive process as it entails block meshing of the parts as compared to free meshing (with mesh controls) in case of a tetrahedron mesh. Moreover, changes in geometry and setup could not be readily re-applied as the entire model had to be re-meshed as a brick mesh.

3.4.6 Interpolating Temperatures From A 4-Noded Mesh To A 10-Noded Mesh

As the brick mesh was not found suitable to readily incorporate changes in geometry, the 10-noded mesh was modified for use in a thermal simulation. The 4 corner nodes in the mesh were used to conduct transient heat transfer simulation. The temperature results were then automatically interpolated to the mid-side nodes of the same second-order mesh. This can be readily achieved with *TEMPERATURE command in ABAQUS.

3.4.7 Achieving Steady State Boundary Conditions

Behavior of a cold die, and a die which has run a few cycles and has reached steady state, were found to be quite different. The thermal simulation was initially run for 30 cycles with a brick mesh. In the final thermal model with 4-noded mesh, a steady state heat transfer step was included before the start of first cycle of the transient analysis. The steady state heat transfer applied an almost uniform temperature gradient of 80 °C, the initial preheat of the insert at the cavity, to 30 °C at the die surface as described in Figure 3.6. This step helped to hasten the approach to almost-steady state like conditions which could be achieved after only 10 cycles. These temperature results were used as the basis of all subsequent parting plane separation comparisons.

Figure 3.6: Steady state heat transfer step used before the start of cycle 1 in transient analysis

3.5 Simulation of Combined Thermal and Mechanical Loads

Once a thermal model was in place, the temperatures at the nodes were outputted just before onset of intensification, the point where maximum pressure spike is presumed to occur. These temperature values were then applied as a temperature boundary condition to the mechanical simulation. The contact separation and pressure values for parting surfaces and the footprint of dies on the platen were compared for various support setups and clamping conditions.

4 CASE SETUPS AND RESULTS

4.1 Introduction

Three different support setups were analyzed for a cold die. These were with the original L-shaped top supports, the case without any top supports, and the modified gusset supports. The cases were analyzed at 1400-ton and 2500-ton clamping forces at the prescribed cavity pressure of 11,000 psi, and at an increased cavity pressure of 16,000 psi. The original L-shaped support, and no support cases were also compared with a cold flush die. These results are detailed in this chapter.

Also analyzed were cases with modified 4" and 7" thick top supports to try and balance the tie bars. Some results of these modified supports are provided, only to test the suitability of tie bar balancing and for comparison only. The simulations with these modified supports were conducted with an old finite element model which was subsequently modified. Hence, only qualitative comparison with respect to tie bar balancing is possible, without any one-to-one quantitative comparison.

Analysis was also conducted with the spacer block behind the ejector box removed. The final result for this arrangement is also included in this chapter.

Various thermal simulations were conducted with the gusset support case as this setup was the solution which was in place at the die caster's facility at the time of the study. For the duration of building the thermal model, different approaches were used to refine the thermal analysis. Only the final results with heat which are relevant are detailed in this chapter.

4.2 Heat Cycle

The process time and heat cycle which was used for running the thermal simulation is shown in Table 4.1 and Figure 4.1. This was based on the die caster's recommendation and could differ from the actual cycle used on the machine. The same process cycle was used for all comparative studies.

Event	Time (Sec)
Dies closing and clamping	4
Injection of molten metal	0.05
From injection to just before die opening	35
Die opening	4
Part ejection	2
Spray	11
Idle time	4

Table 4.1: Process times detailed by duration of events

Figure 4.1: The process cycle shown on a timeline of 60 seconds

4.3 Results With A Cold Die

The parting plane separation and contact pressure results for the die surface were compared without adding thermal loads. These results have been summarized in this section. All separation contour plots indicate values in mm while all pressure plot values are in MPa. Red color indicates maximum values, while blue indicates the minimum values of separation or pressures, as the case may be. Initial simulations were conducted without adequate support pillars on the die behind the cover the ejector inserts. These

pillar supports were subsequently added to the dies as the results were found to be sensitive to these supports.

4.3.1 The Support Setups Analyzed For a Cold Die

A schematic diagram showing the three different kinds of ejector die supports simulated with the cold die is shown in Figure 4.2. The original L-shaped cross-section top supports, shown as Figure 4.2 (a), were the original supports used at the die caster's facility. This type of support exhibited extensive flashing problem at the top and sides of the die. The no-support case Figure 4.2 (b) was not tried on the machine. However, this configuration was simulated to study the effect of removing the top supports from the machine. The gusset shaped support Figure 4.3 (c) was developed at the die caster's facility, based on some recommendations from this study and also based on their indigenous design.

4.3.2 Results for Original L-Shaped Top Support Case For Cold Die

The results for the parting plane separations with the original L-shaped top support are shown in Figure 4.3. The separation values are indicated in mm. Figure 4.3 (a) shows the separation values with 1400 ton clamping tonnage only, while Figure 4.3 (b) shows the separation values after intensification pressure is applied on the cavity along with the clamp load. Figure 4.4 (a) and (b) show the similar parting surface separation plots for case with full clamp of 2500 tons. Figure 4.5 (a) and (b) show the contact pressure plots for 1400 ton clamp at clamp-only, and clamp plus intensification conditions respectively. Figure 4.6 (a) and (b) show similar pressure plots for 2500 ton case.

The parting plane separation plot for clamp only condition show good contact on the insert face (indicated in blue). There is some separation on the die shoe, mostly around the region near the top of the die.

The parting plane separation plots with clamp plus intensification show maximum separation (indicated in red) on the insert, at the region on top of the cavity. The region below, and on the sides of the cavity show almost perfect contact (indicated in blue). The same pattern is evident with both 1400 and 2500-ton clamping cases. This maximum separation on the insert is around 0.38 mm for 1400-ton, while it is reduces to around 0.27 mm for 2500-ton clamp for the cold die. Thus the separation reduces by around 29 % with an increase of clamping by 44 %, but at the cost of increase in contact pressures which are of the order of 200 MPa in localized regions on the die face for 2500-ton clamp. These pressures are concentrated at the bottom of the die shoe. Pressure pattern is visible from the pressure plots (Figure 4.4 and 4.6) that indicate high pressure for area in contact, and low pressures for the open faces.

The maximum separation and pressure values are tabulated in Table 4.2.

(a) Original support

(b) No support

(c) Tower gusset support

Figure 4.2: The setup of the die and the supports on the ejector platen for (a) Original L-shaped top support, (b) No top support, and (c) the gusset support cases

Figure 4.3: Contour plot for parting plane separations for original L-shaped top support and 1400 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.4: Contour plot for parting plane contact pressures for original L-shaped top support and 1400 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.5: Contour plot for parting plane separations for original L-shaped top support and 2500 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.6: Contour plot for parting plane contact pressures for original L-shaped top support and 2500 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

4.3.3 Results for no top support case for cold die

The result for the parting plane separations with no top support case is shown in Figure 4.7 and 4.9. Figure 4.7 (a) shows the separation values with 1400 ton clamping tonnage only while Figure 4.7 (b) shows the separation values after intensification pressure is applied on the cavity along with the clamp load. Figure 4.9 (a) and (b) show the similar parting surface separation plots for case with full clamp of 2500-ton. Figure 4.8 (a) and (b) show the contact pressure plots for 1400 ton clamp at clamp only and clamp plus intensification conditions respectively. Figure 4.10 (a) and (b) show similar pressure plots for 2500 ton case.

With the supports removed, maximum separation is evident on the sides around the insert on the die shoe, for clamp only condition. Insert face is however in good contact (indicated in blue). With clamp plus intensification pressure, the maximum separation on the insert face is on the side and bottom region around the cavity. The region in blue color show almost perfect contact. The same pattern is evident with both 1400 and 2500-ton cases. The contact pressure plots show high contact pressures near the edges while low pressures in the circular region around the cavity, which is clearly an indication of the effect of intensification pressure which is pushing the die faces open.

The separation values are considerably reduced to 0.18 mm and 0.13 mm for 1400 and 2500-ton clamp respectively when compared with the original top L-support case. There is a reduction of around 28 % with increase in clamp from 1400-ton to 2500-ton clamp. The no-support case with 2500-ton gives the lowest parting surface separation for a cold die. The maximum separation and pressure values are tabulated in Table 4.2.

Figure 4.7: Contour plot for parting plane separations for no top support and 1400 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.8: Contour plot for parting plane contact pressures for no top support and 1400 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.9: Contour plot for parting plane separations for no top support and 2500 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.10: Contour plot for parting plane contact pressures for no top support and 2500 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

4.3.4 Results For Gusset Support Case For A Cold Die

The result for the parting plane separations with the gusset support mount case is shown in Figure 4.11 and 4.13. Figure 4.11 (a) shows the separation values with 1400 ton clamping tonnage while Figure 4.11 (b) shows the separation values after intensification pressure is applied on the cavity along with the clamp load. Figure 4.13 (a) and (b) show the similar parting surface separation plots for case with full clamp of 2500-ton. Figure 4.12 (a) and (b) show the contact pressure plots for 1400 ton clamp at clamp only and clamp plus intensification conditions respectively. Figure 4.14 (a) and (b) show similar pressure plots for 2500 ton case.

By comparing separation and pressure plots of no-support case (Figure 4.7 to Figure 4.10) with the gusset support case (Figure 4.11 to Figure 4.14), it is evident that they are indistinguishable. This is true for 1400 as well as 2500-ton clamping tonnages. This was somewhat expected as both supports have very similar load paths though the die face. The no-support case was not tried at the facility, but as a consequence of better simulation results without supports, the new gusset support scheme was built and utilized, to provide better support to the platens. This arrangement was found to give good results with respect to flashing. The gusset support might also help in strengthening the ejector platen. The effect of gusset support on the platen is further discussed in a following section. The maximum separation and pressure values are tabulated in Table 4.2.

Figure 4.11: Contour plot for parting plane separations for gusset top support and 1400 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.12: Contour plot for parting plane contact pressures for gusset top support and 1400 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.13: Contour plot for parting plane separations for gusset top support and 2500 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

Figure 4.14: Contour plot for parting plane contact pressures for gusset top support and 2500 ton clamping tonnage with (a) clamp only, and (b) clamp plus intensification

4.3.5 Comparison Of Results For Proud Versus Flush Inserts For A Cold Die

Figure 4.15 shows the comparison between parting surface separation for a proud (a) and a flush (b) die in cold condition for a 1400-ton clamp and intensification for L-shaped supports. Figure 4.16 (a) and (b) show similar comparison for 2500-ton clamping plus intensification. It is clear that the separation values increase with the flush inserts with L-supports while the die is cold.

Similar comparison plots with the L-supports removed are shown in Figure 4.17 (a) and (b), and Figure 4.18 (a) and (b). Again, the separation increases for the cases with flush inserts. The separation pattern however appears very similar in both the flush and proud dies. The maximum separations with L-supports are still at the top, while the without supports these separations are along the sides of the cavity. As these results are for a cold die, the results for a hot die might differ showing little difference in separation between a proud and a flush insert. This is evident from observations from the field, as well as from previous computer simulations [10].

Figure 4.15: Comparison of parting plane separation plots for (a) proud vs. (b) flush inserts for cold die with original L-shaped top support and 1400 ton clamping tonnage at clamp plus intensification

Figure 4.16: Comparison of parting plane separation plots for (a) proud vs. (b) flush inserts for cold die with original L-shaped top support and 2500 ton clamping tonnage at clamp plus intensification

Figure 4.17: Comparison of parting plane separation plots for (a) proud vs. (b) flush inserts for cold die with no top support and 1400 ton clamping tonnage at clamp plus intensification

Figure 4.18: Comparison of parting plane separation plots for (a) proud vs. (b) flush inserts for cold die with no top support and 2500 ton clamping tonnage at clamp plus intensification

4.3.6 Results for cases with increased cavity pressure

Figures 4.19 to 4.24 show the effect of increase in cavity intensification pressure from the recommended 11000 psi to 16000 psi on the separation values. These results are compared for the three different support structures, both at 1400-ton and 2500-ton clamping tonnages. Across all results, it is evident that the separation values increase dramatically, close to double in most cases. The worst case however is still the lower clamp of 1400-ton with the top supports in place. The maximum separation values for this case are considerably higher at 0.81 mm on the insert for 16000 psi pressure, vs. a value of 0.43 mm for 11000 psi. These results give credence to the fact that flashing may be caused due to considerably high pressures encountered due to dynamic effects of molten metal being pushed inside the cavity. The increased cavity pressures of almost 1.5 times gives an idea of such an effect.

The maximum separation and pressure values are tabulated in Table 4.2.

Figure 4.19: Comparison of parting plane separation plots for (a) 11000 psi vs. (b) 16000 psi cavity pressure for cold die with original L-shaped support and 1400 ton clamping tonnage at clamp plus intensification

Figure 4.20: Comparison of parting plane separation plots for (a) 11000 psi vs. (b) 16000 psi cavity pressure for cold die with original L-shaped support and 2500 ton clamping tonnage at clamp plus intensification

Figure 4.21: Comparison of parting plane separation plots for (a) 11000 psi vs. (b) 16000 psi cavity pressure for cold die with no top support and 1400 ton clamping tonnage at clamp plus intensification

Figure 4.22: Comparison of parting plane separation plots for (a) 11000 psi vs. (b) 16000 psi cavity pressure for cold die with no top support and 2500 ton clamping tonnage at clamp plus intensification

Figure 4.23: Comparison of parting plane separation plots for (a) 11000 psi vs. (b) 16000 psi cavity pressure for cold die with gusset support and 1400 ton clamping tonnage at clamp plus intensification

Figure 4.24: Comparison of parting plane separation plots for (a) 11000 psi vs. (b) 16000 psi cavity pressure for cold die with gusset support and 2500 ton clamping tonnage at clamp plus intensification

4.3.7 Summary of Results With The Cold Die

Table 4.1 gives a summary of the maximum parting surface separation on die shoe and insert, and the contact pressures for the different cases studied for the cold die.

Case #	Clamp (Ton)	Cavity Pressure (psi)	Top Support	Proud or Flush	Max separation on insert (mm)	Max separation on die shoe(mm)	Max pressure on die surface (MPa)
1	2500	11000	Original	Proud	0.27	0.20	259
2	2500	11000	No-support	Proud	0.13	0.09	194
3	2500	11000	Gusset	Proud	0.12	0.09	182
4	1400	11000	Original	Proud	0.38	0.32	88
5	1400	11000	No-support	Proud	0.18	0.12	84
6	1400	11000	Gusset	Proud	0.17	0.12	86
7	2500	11000	Original	Flush	0.32	0.12	252
8	2500	11000	No-support	Flush	0.19	0.02	205
9	1400	11000	Original	Flush	0.43	0.25	54
10	1400	11000	No-support	Flush	0.26	0.08	107
11	2500	16000	Original	Proud	0.51	0.43	226
12	2500	16000	No-support	Proud	0.25	0.15	172
13	2500	16000	Gusset	Proud	0.25	0.17	164
14	1400	16000	Original	Proud	0.81	0.67	Low
15	1400	16000	No-support	Proud	0.37	0.28	80
16	1400	16000	Gusset	Proud	0.37	0.29	81

Table 4.2: Summary of maximum parting surface separation on inset and die shoe and the maximum contact pressures for cases with cold die

4.4 Other Cases Simulated With Cold Die

Apart from the simulations discussed in section 4.2, other cases were simulated for a cold die to gain insight into what was causing the die to flash. Notable amongst these were with modified supports to balance the tie bars, results with die a setup without the support block behind ejector die, and a series of simulations to study the effect of increasing gaps between the original L-shaped top supports. The final results of these are documented here.

4.4.1 Balancing Of Tie Bars

One of the objectives in providing a die which was centered on the platen was to balance the tie bars. In studying the contribution of top supports, smaller size support structures namely 4" and 7" thick plates were used to balance the tie bar reaction forces.

Figure 4.25 shows the position of the original and the modified supports with respect to the die. Table 4.2 indicates the change in parting plane separation with a corresponding change in parting plane imbalance. With the initial top support in place, the tie bars were around +22% out of balance, from the top to the bottom bars. In effect the top two tie bars were being stretched. However, in absence of any support, the balance shifts to – 10%, indicating that without the top supports, the bottom tie bars are being stretched more than the top.

The 4" modified support block was found to balance the loads. However, it did not improve the parting plane separations substantially compared to the L-shape brackets and the lowest parting plane separations were still found with the no-support case. The presence of the support still held the die open and, if implemented, would allow the die to flash. Thus the tie bar balance or imbalance was not found to be a predictor of behavior on the die face.

Supports	Clamping (Ton)	Tie Bar Imbalance (%)	Parting Surface Separation
Original L	2500	22 %	Decreasing Separation
7" Support	2500	7 %	
4" Support	2500	2 %	
No-support	2500	-10 %	
Original L	1400	24 %	Decreasing Separation
7" Support	1400	8 %	
4" Support	1400	2 %	
No-support	1400	-10 %	

Table 4.3: Trends in tie bar balancing compared with reduction in parting surface separation

Figure 4.25: An illustration of the modified supports used to balance tie bars - (a) Original L-shaped supports, (b) Modified support with 7" thick plates, (c) Modified support with 4" thick plates

4.4.2 Results With The Spacer Block Behind Ejector Die Removed

To find out if the spacer behind the ejector die was contributing to the separation on the parting surface, simulations were conducted with the support blocks removed. Without documenting the detailed results, it was found that separation, with and without the spacer were found to be identical. This suggested that spacer was merely transferring the clamp load on the parting surface and had no role to play in affecting the separations.

4.4.3 Increased Gap Between Top Supports For Cold Die

To study the effects of increasing the gap between the top supports to help minimize separations, simulation were conducted by successively increasing the gap value. These results are shown in Figure 4.26 for a 2500-ton clamping tonnage. The plots illustrate that minimum separation was still achieved with no-support case suggesting that any obstruction to the closing of the die, in the form of support on the top held the die open, and would be counter to efforts to minimize separations.

Figure 4.26: Comparison of parting plane separation plots for cold die with 2500 ton clamping tonnage with (a) no top support, (b) L-shaped top support with 1 mm gap, (c) L-shaped top support with 0.5 mm gap, (d) L-shaped top support with no gap

4.5 Results of Parting Plane Separations With Addition Of Thermal Loads

To study the effect of heat on the parting plane separations, simulations were conducted with thermal loads added to the gusset support case. During the initial stages numerous thermal models were constructed as detailed in Chapter 3 on model building. Only the results with the final model are documented here as it was found to be an acceptable result and one that would help in providing sufficient insight into the effect of thermal loads. The results shown are for heat after 10 cycles when almost steady-state like conditions were achieved.

4.5.1 Results With 1400 Ton Clamping For Gusset Support

Figure 4.27 (a) shows the parting plane separation plots for 1400-ton clamp while Figure 4.27 (b) shows the results after application of intensification pressure. The plots show good sealing around the cavity region with the clamp-only condition (indicated by blue). The maximum separations are around the bottom edges of the insert. The die shoe shows perfect contact. With the application of cavity pressure, the separation plot shows distortion of the parting surface with maximum separations on insert lying near the gate area and the top. For the die shoe, maximum separations are close to the gate area.

4.5.2 Results With 2500 Ton Clamping For Gusset Support

Figure 4.28 (a) shows the parting plane separation plots for 2500-ton clamp while Figure 4.28 (b) shows the results after application of intensification pressure. The plot is quite similar to the plot for 1400-ton clamp, showing good sealing around the cavity region with the clamp-only condition (indicated by blue). With the higher clamping tonnage, the extent of the blue region increased. The maximum separations were still found around the bottom edges of the insert while the die shoe was in contact. With the application of cavity pressure, the maximum separations on the insert were along the sides and the top while the maximum on the die shoe was near the gate area. The separation values were lower as compared to 1400-ton clamp case.

4.5.3 Comparison of Separation Results With And Without Heat

Table 4.4 shows a comparison of maximum separation values on the insert for the gusset support case with and without addition of thermal loads. The results were documented for 1400 and 2500-ton clamping tonnages. The results show that parting plane separations appear to increase with the addition of heat.

Clamping	Support	Heat vs. No Heat	Maximum Parting Plane Separations on Insert (mm)
1400	Gusset	No heat	0.143 to 0.164
1400	Gusset	Heat added	0.181 to 0.208
2500	Gusset	No heat	0.100 to 0.122
2500	Gusset	Heat added	0.121 to 0.143

Table 4.4: Comparison of maximum parting plane separation on the insert for gusset support case, with and without heat added

Figure 4.27: Parting plane separation plots after addition of heat for gusset support with 1400 ton clamping tonnage with (a) clamp only and (b) clamp plus intensification

(a)

(b)

Figure 4.28: Parting plane separation plots after addition of heat for gusset support with 2500 ton clamping tonnage with (a) clamp only and (b) clamp plus intensification

4.5.4 Temperature Comparison

Figures 4.29 and 4.30 show the temperature fields comparison between (a) actual thermal graph from a steady state die from GM, and (b) results from thermal simulation after die opening, for ejector die and cover die respectively. The thermal graphs were obtained from an actual die and it was assumed that almost steady state condition was achieved.

The low resolution of the thermal images restricts a detailed quantitative comparison. However as can be seen, the maximum temperatures in the thermal graph are concentrated in and around the cavity region, while the rest of the region on the die is only slight above room temperature. This is also true for the simulation results which indicate a similar thermal field pattern.

The comparisons also indicate that temperatures in simulation results are lower by an average of around 20-30 °C as compared to the thermal graph image. This could be because the simulation results are for 10 cycles which may not yet be at steady state, where as the thermal graphs were assumed to be for almost steady state conditions. Moreover, the simulation assumes perfect cooling lines without clogging, and a constant heat removal of 20% in every spray. These parameters are average values for a die casting process and may change during actual process conditions.

The die caster's recommended melt temperature was 680 °C. However, an initial metal temperature of 650 °C was used which is typical for a casting cycle like this one. It was assumed that some heat is lost during the ladling and pouring of metal. Finally, the location of cooling lines in the FE model was approximate and may cause some difference in the thermal field. In absence of overflows in the finite element model, the regions around the cavity are cooler in simulation as compared to the actual thermal pattern.

Overall the thermal pattern in the actual die appears similar to that in the simulation and provides some validation of the thermal simulation.

(a)

(b)

Figure 4.29: Temperature field for ejector die after steady state conditions (a) thermal graph from actual die in °F, (b) from simulation in °C

(a)

(b)

Figure 4.30: Temperature field for cover die after steady state conditions (a) thermal graph from actual die in °F, (b) from simulation in °C

4.6 Checking For Possible Die and Machine Damage

Initial designs with the original L-supports were to center the die on the platen, as well as to provide additional support. However, one of the concerns was to check the structural integrity of the machine and to study the suitability of the gusset setup. As a consequence platen bending and contact pressures on face of platen for simulations with the three types of supports, namely original L-shaped, no-support, and gusset support cases were compared. Also compared were the values of stresses in die and inserts to check for possible failures. Section 4.7 plots the stresses for the hot die.

4.6.1 Effect of Support On The Ejector Platen

Figure 4.31 shows the comparison of contact pressure plots of the footprint of different support setups, along with the spacer block, on the ejector platen. Figure 4.31 (a) show pressure values for the original L-shaped top supports, (b) for no top support case, while (c) is for the gusset support case. All these plots are for 2500-ton clamping tonnage and are plotted on the same color scale.

The maximum contact pressures occur at the bottom portion of the spacer block with the top L-shaped supports. This location of the maximum pressure shifts to the top region at the spacer block and ejector platen interface when the top supports are removed. The maximum pressure without top support increases to 69.5 MPa where as it was 42.7 MPa with the top supports in place.

Finally, with the gusset support in place, the contact pressures appear to be distributed on the entire face and are much lower in value of 26 MPa. This strengthens the suitability of the gusset arrangement in helping to distribute the clamp load more uniformly over the face of the platen and avoiding high contact pressure at the support footprint on the ejector platen.

The maximum contact pressure values are tabulated in Table 4.5

Top Support	Max Contact Pressure (MPa)
Original	37.4 - 42.7
No-support	64.1 - 69.5
Gusset	21.4 - 26.7

Table 4.5 Contact pressure plots of footprint of spacer support on ejector platen for the three support setups

Figure 4.31: Contact pressure plots of footprint of spacer support on ejector platen for (a) original L-shaped top support (b) no top support and, (c) gusset support

4.6.2 Effect of support setups on bending of platens

Figure 4.32 shows a schematic of bending of platens for the different support arrangements. These plots are magnified 100 times for better illustration. The green indicates the distorted setup while the dark red indicate the original or undistorted setup.

As evident from Figure 4.32 considerable movement was observed in the ejector platen which is free to slide in the Z-direction, or the direction along the tie bars, as well as move up along Y-direction. There were some differences in the pattern of this movement for different setups. For the case with no-support and gusset support, the ejector platen appeared to bend inwards from the top while the whole platen appeared to rise up along Y-direction with clamp force.

The ejector platen in case of the original L-shaped supports appeared to bend inwards, but this time towards the bottom. This provides an indication of the considerably high stiffness of the L-supports in restraining the platen movement.

The cover platen by contrast was relatively lacking in displacement which was quite evident in ejector platen. Moreover, there appeared to be little difference between the movements of cover platen for the three setups. This could be because of the restrictive boundary conditions used to secure the cover platen at the bottom face.

This movement can be quantified from Figures 4.33 and 4.34 which plot the movement along Z-direction for cover and ejector platens respectively. For the cover platen (Figure 4.31), the no-support case (a) and the gusset support (c) appeared identical with maximum displacement in the middle indicated by red color. The case with L-shaped support (b) shows the imbalance in movement with the top showing more displacement than the bottom.

The contour plot for ejector platen in Figure 4.34 showed this imbalance more clearly. Maximum movement of around 5.5 mm appeared at the top of the platen for the case without support. This movement was found to be quite less for the case with L-support but the location of displacement was shifted to the bottom portion. With the gusset support, the maximum movement was still at the top; however the magnitude of this displacement was less at 4.7 mm when compared with the no-support case

The Z-displacement plots for the ejector platen indicate that the gusset support contributes in stiffening the ejector platen and preventing its unusual bending.

Platen	Support	Max Displacement along Z (mm)	Location of Max Displacement
Cover Platen	No-support	3.20 - 3.29	Center
	Original	2.90 - 3.01	Top
	Gusset	3.20 - 3.29	Center
Ejector Platen	No-support	5.23 - 5.51	Top
	Original	3.69 - 3.87	Bottom
	Gusset	4.78 - 4.96	Top

Table 4.6: Maximum displacement along Z and its location for the cover and ejector platen

Figure 4.32: Schematic showing the movement of ejector platen with different top support structures, (a) no-support, (b) original L-shaped support, and (c) gusset support

Figure 4.33: Contour plots showing movement of cover platen along Z-axis which is along the tie bar or along the direction of application of clamp, with different top support structures, (a) no-support, (b) original L-shaped support, and (c) gusset support

Figure 4.34: Contour plots showing movement of ejector platen along Z-axis which is along the tie bar or along the direction of application of clamp, with different top support structures, (a) no-support, (b) original L-shaped support, and (c) gusset support

4.6.3 Checking For Plastic Deformation of Die Components

With concerns for coining of die components with high clamping of 2500-ton, stresses obtained from simulation for cold die were checked and compared with the typical yield strength values for 4140 steel and H-13 steel. Even localized stress higher than, or close to yield strength value would be a clear indication of plastic deformation similar to forging of dies. This comparison does not involve effects of thermal and mechanical fatigue over many cycles.

Table 4.7 shows the maximum equivalent Mises stresses obtained with different components like the die, support block and the platens for various cases.

Support Case	Cavity Pressure	Ejector Die	Ejector Insert	Cover Die	Cover Insert	Support Block	Ejector Platen	Cover Platen
	psi	MPa	MPa	MPa	MPa	MPa	MPa	MPa
Original	11000	242.4	164	235.3	148.1	280.9	60.8	264
No-support	11000	338.7	180.6	334.3	136.1	188	89.81	243.8
Gusset	11000	324.3	180.3	344.6	132.8	299.9	59.86	245.4
Original	16000	323.7	236	238	173.1	284.6	60.07	262.3
No-support	16000	366	255.6	338.6	161.1	186.4	90.06	240.7
Gusset	16000	371.12	255.2	349.3	160.9	298.4	60.03	242.3

Table 4.7: Maximum equivalent (Mises) stresses obtained from simulation results for cases with 2500-ton clamp for cold die

As seen from Table 4.7, maximum stresses are obtained with no-support and gusset support cases. The values of yield strengths typical for steels are shown below:

Yield strength for 4140 Steel (normalized) = 675 MPa [2]

Yield strength for H13 Steel (air or oil quenched) = 1650 MPa [2]

The maximum stress values do not exceed the yield strength values shown below. A cavity pressure of 16000 psi would be an approximation of the dynamic cavity pressure conditions. The stresses with 16000 psi also do not exceed the yield strength indicating a good margin of safety for plastic deformation of die.

The equivalent stresses for the cover die and the ejector die for a cold die are plotted in Figures 4.35 and 4.37, while Figure 4.36 and 4.38 are the maximum principal stresses respectively.

As can be seen from these figures, the maximum stresses are concentrated around the corner nodes which are tied to the corner nodes of adjacent component in the assembly, a technique used to model the effect of an assembly fastener. This means that these stresses may be unnatural with stress values that are quite high as compared to normal stresses encountered during actual conditions. A yield strength check would still be feasible as it would indicate the worst case situation.

Comparison also shows that average stresses are higher in the ejector die and mostly concentrated around the top pillars. Figures 4.39 and 4.41 show similar Mises stress plots for the ejector insert and cover insert, while Figure 4.40 and 4.42 are the maximum principal stresses respectively.

Equivalent stresses in both the inserts appear to be the similar. However, the principal stress plots indicate that stresses are higher on the cover insert. As both the plots are on the same color scale, the maximum on cover side ranges from 70 to 100 MPa, while on the ejector insert, the maximum is up to 70 MPa. Similar values for the dies are 100 to 125 MPa for the cover die and 75 to 100 MPa on the ejector die.

This might suggest higher chances of breakage on the cover insert, which was also found at the die caster's facility. Results for the hot die might differ from the ones for the cold die.

Figure 4.35: Equivalent von Mises Stresses in Cover Die for Cold Die, Gusset Support with 2500-ton Clamp

Figure 4.36: Max Principal. Stresses in Cover Die for Cold Die, Gusset Support with 2500-ton Clamp

Figure 4.37: Equivalent Mises Stresses in Ejector Die for Cold Die, Gusset Support with 2500-ton Clamp

Figure 4.38: Max Principal. Stresses in Ejector Die for Cold Die, Gusset Support with 2500-ton Clamp

Figure 4.39: Equivalent Mises Stresses in Ejector Insert for Cold Die, Gusset Support with 2500-ton Clamp

Figure 4.40: Max Principal Stresses in Ejector Insert for Cold Die, Gusset Support with 2500-ton Clamp

Figure 4.41: Equivalent Von Mises Stresses in Cover Insert for Cold Die, Gusset Support with 2500-ton Clamp

Figure 4.42: Max Principal Stresses in Cover Insert for Cold Die, Gusset Support with 2500-ton Clamp

4.6.4 Checking For Supports Behind The Inserts

Figure 4.43 show the contact separation plots between the ejector insert and ejector pocket (a) and the cover insert and cover pocket respectively for the warm die. Figure 4.44 show the respective contact pressure plots for the same case.

The separation plots are an indication of support which the die pocket provides behind the inserts. A less support might be an indication of greater bending in those areas. The contact separation plots indicate fairly good contact without much separation. However, there is very small separation on the top edge of the cover insert.

The contact pressure plots show a much clearer picture. For the cover side, the maximum pressures are concentrated around the top region on the back side of the insert. In case of ejector insert however, the maximum pressures are at the bottom and the center and are better distributed. This might indicate a better support for the ejector insert as compared to the cover insert.

(a)

(b)

Figure 4.43: Contact Opening between die and insert (a) ejector side (b) cover side, 2500-ton, 11000 psi cavity, gusset support

Figure 4.44: Contact Opening between die and insert (a) ejector side (b) cover side, 2500-ton, 11000 psi cavity, gusset support

4.7 Additional Analysis of Stress and Displacement²

The purpose of this analysis is to clarify some of the inconsistencies in the simulation results obtained from the runs on the ABAQUS version 5.8. Due to convergence problems on the new version 6.3 and time constraints, the stress results were obtained by running a dummy step using displacements from the .dat file of the 5.8 version. The results however were not as expected. Tensile stresses were observed on the cavity face of the cover insert while compressive stresses were expected and hence it was decided that the analysis be rerun in the new version 6.3.

The geometry of the model is greatly simplified compared to the actual die cavity, particularly with respect to die cavity details. This means that stress risers that might be actually present in the insert are not captured in the model. The other quality is that the meshing could be a little coarse for accurate stress.

The analysis was performed based on an elastic stress-strain response due primarily to the fact that we used the same model setup as used previously for parting plane separation analysis. Plastic behavior was not modeled. The elastic analysis can over estimate stress magnitudes particularly if the stresses are high (near yield). The stresses are much lower in our case, so we should not have that problem.

Our best guess as to the root cause of high stresses is thermal growth and crowning of the insert coupled with the high clamping tonnage. Our temperature analysis probably does not accurately represent tip cooling or the effects of spray in this region of the die. Obviously, there are limits to the interpretations that can be drawn from the model, but the trends should be meaningful.

Low cycle fatigue failure is normally associated with tensile stresses. The scales on the stress plots are set so that the compressive or negative values are displayed in black. This allows the tensile stresses to be observed more clearly. The units for stress in all cases are MPa.

4.7.1 Stress Results

The following figures are snap shots captured at intensification. The general trend observed is that intensification occurs at about 20 milliseconds after injection. The simulations hence have been run to observe the stresses at this point since the problem of flashing would be the greatest here. In terms of the model, the die must be at its coolest and both clamp and intensification loads are applied. The temperatures are approximately quasi-steady state for the point in the cycle.

Figure 4.45 shows the maximum principal stresses for the cover insert with a hot die. Majority of the cavity side is in compression and the back side in tension. The dark blue regions are in the 0-40 MPa range and the light blue regions at the centre of the back side which is the area of maximum tensile stresses are about 115-150 MPa. The high tensile stresses at the corners near the shot block and the compressive stresses near the corners on the back side are probably due to the restraint caused by the stiffer die. The tensile stresses on the sides are due to the buckling action produced by the cavity pressure.

² This analysis was performed by Jeeth Kinatingal

In contrast, the cold die stresses are shown in Figure 4.46. The loads for the cold die case are clamping, intensification and heat but the initial temperature of the die is uniform and the die is not in quasi-steady state. The front side is mostly in compression and the back side in tension with the maximum tensile stress in the range of 60-90MPa.

Figure 4.47 shows the maximum principal stresses in the ejector insert with a hot die. As in the case of the cover insert here also we see compressive stresses on the cavity side and tensile stresses on the back side. The maximum tensile stress is about 95-125MPa. Figure 4.48 is the cold die version of the ejector insert which shows stress magnitudes similar to the cover side; the cavity side is in compression and the back side in tension with the maximum tensile stresses in the 50-80MPa range.

The stresses in the holder blocks are much lower than in the inserts as might be expected. Figure 4.49 shows the maximum principal stresses in the cover holder block for the hot die case. As expected, the cavity side is in compression and the back side in tension with the maximum tensile stresses between 25-50MPa. Figure 4.50 shows the stresses for the cold die case. Here we see that the maximum stresses are higher, 85-130MPa. This is because the dies grow inwards with heat and therefore the thermal loads reduce the tensile stresses caused on the back side by the mechanical loads. As a result there is also a larger area on the cavity side which is under compression. We also see that the stresses are highest on the back side in the region that is not supported by the pillars.

Figures 4.51 and 4.52 show the maximum principal stresses on the ejector holder block for the hot and cold die cases respectively. The stress patterns are similar to the cover holder block with the cavity side being in compression and the back side in tension. The maximum stresses are 43-54MPa with the hot die and 43-57MPa with the cold die. The reduction in the tensile stresses on the cavity side with heat is not as prominent as in the cover holder block. This is because of the much larger thickness of the ejector holder block which reduces the thermal effect on its back side. However, as in the cover holder block, here also we see a larger area of the cavity side in compression as compared to the cold die case. This again is due to the thermal growth of the die. The stresses are higher near the top edge of the block. This is because the dies are not oriented centrally between the tie bars and the platens but a little lower as result of which the support is lesser for the top half causing more bending and thereby higher stresses in this region.

The displacement of the inserts is shown in profile in Figure 4.53. The deflection pattern can be viewed clearly in this figure especially in the top view. The effect of the mechanical loads (clamp and intensification) is more than that of the thermal loads as a result of which we see the inserts buckling outwards. The cover insert has more displacement as compared to the ejector insert as it is located within a holder block of much lesser thickness (stiffness). The ejector insert is somewhat thicker than the cover and therefore is a little stiffer than the cover insert. Also, the distribution of the pillars on the ejector side is a little better than on the cover side (see Figure 4.54 - a schematic showing the location of the cooling lines and pillars with the pillars denoted by rectangular cross sections.)

The displacement of the cover die and the ejector die are shown in figures 4.55 and 4.56 respectively. We see lesser displacement in the ejector die as it is stiffer due to its larger thickness. Also, the maximum displacement in the cover die is seen clearly in the area that is not supported by pillars.

Figure 4.57 is a parting plane separation plot of the inserts (on the left) and the dies (on the right) for hot die with clamp loads only (Figure 4.57a) and both clamp and intensification loads (Figure 4.57b). We see that the inserts must be a little thicker around the cavity and at the biscuit resulting in a crown shape that peaks at the cavity edge and drops down toward the edges. When the machine is clamped, the inserts come in contact at the dark blue regions corresponding to the top of the crown and the insert is compressed into the die block until the entire die takes up the load (Note that most of the die separation plot in figure 4.57a is dark blue). With intensification, Figure 4.57b, the insert moves some more, but the dies remain mostly in contact. The separation pattern suggests a folding-like movement with the center top and entire bottom edge of the cavity moving the most. The movement tendency is very clear in the displacement plot shown in Figure 1.53. Comparing figures 4.57 (b) and 4.54(a) (see the pillar location) we see that the parting plane separation is more in the regions where there are less support from pillars.

Figures 4.58 and 4.59 show the contact pressure plots for hot and cold dies respectively with clamp loads only (Figures 4.58a and 4.59a) and clamp and intensification loads (Figures 4.58b and 4.59b). First, we have only the clamp loads (Figure 4.59a). Since the inserts are proud, the die surfaces do not come in contact and we have a uniform distribution of the load across the insert parting surface. Next when the thermal loads are applied (Figure 4.58a), the insert parting surface would seem to take on a crowned shape and this concentrates the load around the cavity including the biscuit and runner area. In both cases (hot die, Figure 4.58b and cold die, Figure 4.59b) the inserts are shoved into the holder block when the intensification pressure is applied until the block itself helps to distribute the loads over a larger surface. The contact pressures are higher near the top edge of the block. This again is because the dies are not oriented centrally between the tie bars and the platens but a little lower as result of which the support is lesser for the upper half.

As discussed previously, there is not much difference in the stresses on the two inserts when cold. The differences between cold and hot in the die basically show the effects of load path changes. Judging by eye, if pillar locations are placed on the stress plots of the inserts, the high stress regions falls between the pillars. We know this is true for the holder block from figures 4.48 and 4.49 but we don't know if it is the pillars or the cooling pattern that is more significant.

4.7.2 Summary

Many of the inconsistencies that were found in the previous analysis have been resolved. We can make the following conclusions from these results.

- The lack of cavity details makes conclusions speculative.
- The tensile stresses are highest on the back side of the cover insert in the region that is not supported by the pillars.
- The thermal loads reduce the tensile stresses caused on the back side by the mechanical loads.
- There is a change in the load path with heat. The shoe and the region around the shot hole of the die take much of the clamp load.
- The reduction in the tensile stresses on the cavity side of the ejector holder block with heat is not as prominent as in the cover holder block. This is because of the much larger thickness of the ejector holder block which reduces the thermal effect on its back side.
- The stresses and the contact pressures are higher near the top edge of the block. This is because the dies are not oriented centrally between the tie bars and the platens but a little lower as result of which the support is lesser for the top half causing more bending and thereby higher stresses in this region.
- The cover insert has more displacement as compared to the ejector insert as it is located within a holder block of much lesser thickness (stiffness). The ejector insert is somewhat thicker than the cover and therefore is a little stiffer than the cover insert. Also, the distribution of the pillars on the ejector side is a little better than on the cover side
- With a hot die, the area surrounding the cavity is in contact at clamp, but opens somewhat with pressure. This appears to be due at least in part because the cover side holder block allows the insert to move and twist slightly. This motion might be reduced with a stiffer cover insert, modified pillar locations, or improved cooling that will help to manage the thermal growth that is affecting the load path.

Figure 4.45 - Max Principal Stresses - Cover Insert - Hot

Figure 4.46 - Max Principal Stresses - Cover Insert - Cold

Figure 4.47 - Max Principal Stresses - Ejector Insert - Hot

Figure 4.48 - Max Principal Stresses - Ejector Insert – Cold

Figure 4.49 - Max Principal Stresses - Cover Die - Hot

Figure 4.50 - Max Principal Stresses - Cover Die - Cold

Figure 4.51 - Max Principal Stresses - Ejector Die – Hot

Figure 4.52 - Max Principal Stresses - Ejector Die - Cold

Figure 4.53 - Insert Displacement Pattern (cover – ejector)

(a)

(b)

Figure 4.54 - A schematic showing the location of pillars and cooling lines on (a) Cover die and (b) Ejector die

Figure 4.55 – Cover die displacement pattern

Figure 4.56 – Ejector die displacement pattern

Figure 4.57 - Parting plane separation plots, Hot Die (a) clamp only and (b) clamp plus intensification

Figure 4.58 - Contact pressure plots, Hot Die (a) clamp only and (b) clamp plus intensification

Figure 4.59 - Contact pressure plots, Cold Die (a) clamp only and (b) clamp plus intensification

5 Modeling Summary and Conclusions

During the course of this study, a lot of time and effort was spent in trying to understand how best to model the problem without over estimating, or under estimating the boundary conditions and restraints used. A part of this effort was to build a realistic thermal model. As numerous assumptions were made throughout the analysis, it was important to understand their limitations.

The finite element study did provide good insight in studying die casting die distortions for a real industrial die. Even though the process of building and modifying the model was time intensive, it was certainly easier than making a similar change on the machine. This fact was quite evident from the results obtained for numerous scenarios like no-support cases, increased cavity pressure, modified supports to balance tie bars. Most of these scenarios could not be tried at the die caster's facility but their results were invaluable in suggesting alternative designs.

As is true for any work, the challenges faced were actually opportunities to learn more about finite element studies. A good example is the apparent inability and subsequent delay in solving the thermal problem to satisfaction because the 10-noded element was found to be unsuitable for thermal studies. It is likely that there still are problems with the stresses due to the 10 noded elements

Finally, the study also served as a good validation for the general procedure used at the Ohio State University for modeling die casting process.

The finite element model was successful in modeling an actual die casting process for an industrial die. The model was also successful in modeling the die and machine assembly of components. Some of the modeling techniques were used for the first time in the OSU group. Such as the beam element leader pins and the simulation of the tie nut as a shell element. These techniques were useful in adding sufficient die and machine geometry detail to the model while maintaining tractability.

Right from the start it was clear that the top L-shaped supports did not help in minimizing the parting plane separations. They were in fact contributing to flash, even though these results were from the cold die. The results suggested that even though the top supports appeared to balance loading on the die, they were holding the die halves open, enough for them to flash from the top and sides. The results also suggested that to minimize separations, the clamping force should reach the die face, which was clearly not happening with a top support in place.

The gusset support case was found to give good results at the die caster's facility. However, from the results of the simulation, it was evident that the gusset support and no-support cases gave virtually the same values of separations and contact pressures at parting surfaces. This could suggest that the no-support case would work just as well as the gusset. However, with concerns with the structural integrity of the die casting machine in mounting a small die low on a big platen, it was necessary to further investigate effects of using a gusset.

Throughout the course of the study, higher clamping of 2500-ton consistently provided better results for parting plane separations. However, this was at the cost of considerably high contact pressures on the die faces. These high contact pressures not only made simulation convergence difficult, the higher forces did not achieve a corresponding reduction in separation. An increase in clamping force of 72% caused a reduction in separation of around 32%. The higher clamp in effect was jamming the entire structure more tightly, possibly risking damage to the die halves.

The gusset supports achieved better results in the field as far as flashing was concerned. More importantly, the gusset supports probably helped reduce ejector platen bending. The gusset also helped distribute the contact forces more uniformly on the face of the platen and reduced high contact pressures.

Counter to intuition, the balancing of tie bars did not give the best results as far as die parting separations were concerned. Even though the tie bars were almost balanced with 4" supports at the top of the dies, they did not yield minimum separations and the best results were still achieved with no-support at the top. This suggested that tie bar balancing is not a good indicator of controlling parting surface separations when the die is not centered and loads not symmetric.

The results of mechanical simulation with thermal loads indicate a slight increase in maximum parting plane separation for the gusset support case. The maximum separations occur along the sides and the top which was as observed by the die caster. The separation pattern indicates sealing around cavity and the shot block region for a hot die before application of intensification, and a subsequent opening around the cavity when cavity pressure is applied. However even after cavity pressure is applied, the shot block region appears to still be in contact. This non-uniform separation might suggest that the die is likely to flash more as it gets hot. This was also supported by observations made by the die caster. The separation predictions might be slightly different with a spring displacement boundary condition as heat growth does not affect the clamping load with a pressure condition. The simulated conduction represents adjustment of clamping load with growth in die height, an adjustment that was not made in practice.

The tight sealing around the biscuit region is possibly due to uneven heat growth on the parting surface. The insert grows more around the shot block causing the inserts to be held apart at these regions. The uneven heat growth could possibly be due to an overestimation of biscuit temperatures or an inability to effectively model the coolant spray at the shot regions keeping the biscuit hotter than it normally would be.

As the cavity pressure pushes in, the region around the cavity tends to open up even more. This is compounded by the fact that the region behind the inserts is the least supported area allowing for movement in Z direction. This might also suggest the inadequacy of the existing pillar supports in supporting the die.

The high contact pressures at the parting surface and the footprint of die support on ejector platen along with the bending of platen, would suggest unnatural strains on the machine. The machine was designed for big dies centered on the platen. These strains could suggest possible damage and breakage on the machine. The small die was not designed for high clamping forces and may fail due to high localized stresses. Some of these concerns, like die coining, were already shared by the die caster.

Even though checking for die and machine failure was not the primary objective of the study, it was useful to look at the stresses to see if they can provide some insight into possible damage. The die caster was already experiencing failures, mainly on the cover insert. The stress results in chapter 4 for the hot die show slightly higher principal stresses for the cover die which might explain the cause of such failures. The dies appear mostly in tension while the inserts are mostly in compression which is expected. The regions in and around the cavity are in tension due to the heat growth. These stresses are uneven due to the placement of pillars and possibly due to uneven heat growth due to presence of cooling lines. The stress results may be noisy as they were recalculated using displacement values at the nodes.

5.1 Dynamic Effects of Shot-End Power and Fatigue

The scope of this study is limited as it could not adequately account for dynamic effects of the die casting machine. Some insight into the dynamic effects could be achieved from results for the increased cavity pressure of 16000 psi. However, the big machine with considerably high moving masses could only be simulated with a full-fledged dynamic study. The flashing problem observed on 2500-ton machine could very well be due to the higher impact spike and dynamic effects than those experienced on the smaller machine.

The model does not account for thermo-mechanical fatigue of die and machine components. The stresses in dies and inserts are below yield values. However, they are high enough to make fatigue life an issue. This will be of importance if the die failures are to be studied in detail.

6 Experimental Verification of Load Pattern

6.1 Introduction

Building simulation models for casting processes has increased for the last two decades due to the increase in the available computational powers. Finite element and finite difference methods are the main tools to build, solve and analyze simulation models that represent the different casting processes. Though the availability of computational powers makes it possible to run very comprehensive casting models, this is not always the case. The usual technique in building these models is to concentrate on few process characteristics and factors and assume -or neglect- the rest. So approximations and/or assumptions in the solid model, loads definition, process sequence, material physical and mechanical properties, etc. are very common.

Several reasons stand behind this attitude. One important reason is to get a cost effective solution even with sacrificing some generality and/or accuracy of the model. Another reason may be the lack of material properties, especially at high temperatures. The lack of data may also extend to the boundary and initial conditions that should be included in the model. These boundary and initial conditions should represent the working conditions and environment which are subject to change tremendously during different production sessions in the shop floor. For example the cleanness of die cooling lines in a die casting die can affect the heat transfer coefficient by order of magnitude. These approximations and/or assumptions force the modeler to verify his/her simulation model results against experimental measurements.

For more than ten years the Center for Die Casting at the Ohio State University has been extensively involved developing models that predict die deformation in operation. The purpose of the experimental work described in this report is to confirm that the modeling is adequate and can be trusted. The experiments were run on a 250 metric ton Buhler die casting machine available at the Ohio State University. In total, 71 sensors (35 load cells, 32 strain gauges 4 thermocouple) were mounted on the machine in order to dynamically measure three types of quantities:

- Contact forces between dies and platens
- Strains in the tie bars and dies
- Temperatures in the dies

6.2 Experiment Methodology

Two types of loading affect the machine parts and are considered in the simulation model. The first type is the mechanical loads that result from the machine clamping and the applied intensification pressure. The second type is the thermal load that comes from the heat transfer through the interfaces of casting/inserts inserts /dies and inserts & dies/cooling lines and the heat dissipated to the environment.

Load cells were inserted between the dies and platens to measure the contact forces between them on both cover and ejector sides. Special fixtures were designed to hold the load cells. The simulation model was modified to include the load cells.

The reactions in the machine were measured using uniaxial strain gages on the tie bars. Four strain gages were mounted on each tie bar and the average value was used to compare with simulation results. The die casting machine has the capability to give the total clamping load, but, unlike some die casting machines, it does not have the capability to give the load in each tie bar separately. Another important benefit from measuring the strains in the tie bars, besides comparing with the model predictions, was to assure that the tie bars are balanced and the clamping load is symmetric around the vertical axis that passes through the center of the die, as it was assumed in the model.

Strain gages were also used to measure the strains in the dies. Uniaxial strain gages were mounted on the die steel –both cover and ejector sides- to measure the strains in the three directions (X, Y and Z).

Thermocouples were used to measure the temperature in the dies. The goal was to verify the results from the thermal simulation models.

The experiment is designed for two die sets. The first die set, which is covered in this report, is a small die that covers 40% of the platen area. The second die set is a big die that covers 90% of the platen area. The experiments using the second die set is not conducted yet and will be covered in a separate report.

6.3 Loading Conditions

Two different load conditions were applied during the experiment, as shown in Table 6.1. In the first load condition, only the clamping load was applied (in other words there was no actual casting process). This loading condition was very important for verifying the simulation model. It was also very useful in testing our equipment and designs. One important feature in this load condition is that, the clamping force is completely controlled by the machine controlling unit, and hence there were no unpredicted factors that may affect the comparison between the simulation model results and the sensors readings. Under the first loading condition, two clamping loads were examined, 1500 KN, which is the minimum clamping load allowed by the machine, and 2500 KN, which is the clamping load that would be used with our casting.

In the second load condition, an actual casting process was run and several castings were produced. Semi-automatic casting cycle was selected. In this type of cycle the die closing period and casting ejection period are controlled by the machine controlling unit, while the pouring, extracting and spraying periods are controlled by the operator. This cycle was selected because the pouring robot, spraying reciprocator and extractor did not work properly and manual pouring and spraying were used.

Several factors affect the results under the second loading condition. Some factors are completely controllable (for example: the clamping force, and die closing period). Other factors are partially controllable (for example: pouring temperature, spraying time and amount and overall cycle time). The cycle time is given in Table 6.2¹.

Load Condition		Active Sensors
Load Condition I	1500 KN Clamping	Load Cells & Strain Gages
	2500 KN Clamping	Load Cells & Strain Gages
Load Condition II	2500 KN Clamping + Casting	Load Cells, Strain Gages & Thermocouples

Table 6.1: Loading conditions

- 1- Osborne, M. A. (1995), Analysis of the Effects of Die Casting Process Control on Casting Dimensional Variability, Ph.D. Dissertation, Ohio State University.

Cycle Segment	Time (Sec)
Ejector retract	1.5
Machine close	2.5
Machine lock	2.5
Pour charge	2.5
Metal injection	0.5
Solidification	8.0
Decompression	0.5
Machine open	3.0
Ejection	3.0
Casting removal	10.0
Spray release	4.5
Delayed closing	1.5
Total cycle time	40.0

Table 6.2: Cycle time

6.4 Load Cell Placement

Diaphragm, strain gages, load cells were used to measure the contact force between the dies and platens. Eighteen load cells were inserted on the cover side and seventeen load cells were inserted on the ejector side. Each load cell capacity is 445 KN (100,000 lbs). Specifications of the load cells are given in Appendix A

As shown in schematic drawing, Figure 6.1, three steel plates were used, on each side, as a fixture for the load cells. Plates 1&3 were used to protect the platen and die, respectively, from the load cells footprints. Plate 2 was used to restrain the load cells from vertical and horizontal movement. As noticed in the figure, the thickness of plate 2 is less than the height of the load cells to allow the load cells to respond freely to the applied forces.

Plate 1 is clamped to the platen using standard die clamps. Plate 3 is bolted to the die and plate 2 is bolted to plate 3. Figures 2-a and 2-b show the load cells fixture plates on cover and ejector sides respectively. Detailed drawings of these plates are given in Appendix B. Photos of the load cells setting are given in Appendix C. Figure 6.3 shows the load cells location and identification number in both cover and ejector sides.

Flatness of the fixture plates is very important to have good contact between the plates and the load cells. Using a CNC milling machine to manufacture the fixture plates achieved flatness of 0.004 inch in the plates, but at the first run it was clear that this flatness was not good enough. The plates were then ground using a Blanchard Grinding Machine modifying the flatness to 0.001 inch. Even with this level of flatness we had some problems with contact. Two load cells were shimmed –E16 & C15– because they did not have a good contact with the plates. This problem with contact was noticed because the readings from these load cells were very low compared to the neighbor load cells. A feeler gage was then used to check if there is a gap between the load cell and the fixture plates. Depending on the gap size, an appropriate shim was inserted between each load cell and the plate. Although the load cell C13 also needed shimming, it was not shimmed because it was not accessible after installing the dies on the machine.

Figure 6.1: Schematic drawing of the die/platen/load cells setting

(a)

(b)

Figure 6.2: Load cells plates at: (a) cover side & (b) ejector side

Figure 6.3: The load cells identification numbers

6.5 Strain Gages Placement

Uniaxial strain gages were attached to the machine tie bars to measure the longitudinal strains. Four strain gages were attached to each tie bar. Figure 6.4 shows the tie bars identification numbers with respect to the back of the cover platen and the locations of the strain gages on each tie bar. The strain gages were mounted on the tie bars at a distance of 267 mm (10.5 in) from the cover platen. At this distance the strain gages are in the middle between the two platens when the die is closed.

The strain gages were attached to the tie bars using a special type of glue. Specifications of strain gages are given in appendix A.

Figure 6.4: Tie bars identification numbers with respect to the back of the cover platen

A total of 16 uniaxial strain gages were attached to both cover and ejector dies. Figures 5 and 6 show the locations of the strain gages in the cover die and ejector die respectively. There are two strain gages at each location. Table 6.6.3 summarizes the strain gages directions at each location.

Figure 6.5: Locations of strain gages on the cover die Side view

Figure 6.6: Locations of strain gages on the ejector die

Location	Strain gages directions	Location	Strain gages directions
CD1	Y&Z	ED1	Y&Z
CD2	Y&Z	ED2	X&Z
CD3	X&Z	ED3	Y&Z
CD4	Y&Z		
CD5	Y&Z		

Table 6.6.3: Strain gages directions at cover and ejector dies

6.6 Thermocouples Placement

Four K-type thermocouples were used to measure the temperature in four locations in the inserts. Three of them were on the cover side and one on the ejector side. The thermocouples were inserted at a distance of 12.5 mm (0.5 in) from the cavity surface.

The thermocouples were located in four existing holes in the cover and ejector inserts. Figure 6.7 shows the thermocouples locations in the cover and ejector inserts (denoted by T1 – T4). Special H 13 steel fixture was designed and made to mount each thermocouple. The fixture drawing is shown in Appendix B. Each thermocouple was cemented to the fixture using special high temperature cement. Specifications of the thermocouples and cement are given in Appendix A.

Figure 6.7: Thermocouples locations in the cover and ejector sides (dimensions in mm). The thermocouples were located at a depth of 12.5 mm (0.5 inch) from the cavity surface

6.7 Data Acquisition

A PC based data acquisition system was built to collect the data from the transducers. Figure 6.8 shows the main components of this system. Specifications of the signal conditioning and data acquisition & analysis (DAQ) device are given in Appendix A.

Figure 6.8: Elements of a PC based data acquisition system

The total capacity of the system is 88 analog input channels divided into two groups:

- 56 channels for load cells and strain gages interchangeably.
- 32 channels for thermocouples.

The maximum allowable sampling rate of the system for all channels is 333KS/s. The sampling rate used in the experiments was 1KS/s for each channel. The system does not allow different sampling rates for different channels.

The data from all the channels were filtered. The load cells and strain gages channels were programmed with a lowpass filter of 10HZ. While the thermocouples channels are filtered at a nonprogrammable, preset, value of 10KHZ.

Load cells were connected using a full bridge configuration while strain gages were connected using quarter bridge configuration.

A Labview program was written to control the data acquisition system. The main window and diagram window of the program are given in Appendix D.

7 Experiment Results

7.1 Load Condition I – Clamp Loads Only

As mentioned before two clamping loads were applied in load condition I, these are 1500 KN and 2500 KN. The machine applies the load within preprogrammed tolerance. Due to this tolerance the machine clamping load readings were 1575 KN and 2452 KN. At each clamping load twelve runs were conducted. The average of the twelve runs was used to compare sensors readings with the simulation predicted values.

7.1.1 Load cells

Table 7.1 shows the average of the readings and the standard deviation for each load cell at the two clamping loads. Table 7.2 gives the load cells loading predicted by the simulation model. Examining the value of the standard deviation for the load cells readings show that its value is very small compared to the average value. This shows that the repeatability of the measurements is high.

Figures 7.1 and 7.2 show comparisons between the simulation predictions and experimental measurements at a clamping load of 1575 KN for load cells at cover side and ejector side respectively. Figures 7.3 and 7.4 show comparisons between the simulation predictions and experimental measurements at a clamping load of 2452 KN for load cells at cover side and ejector side respectively.

To understand the Figures 7.1-4 one must recall that the die is symmetric around a plane that passes through the center of the shot sleeve hole and perpendicular to the parting plane. The symmetry is very clear in Figure 6.7. Due to this symmetry, only half of the die casting machine was modeled and, as a consequence, the similarity of load cells readings around the symmetry plane was assumed. Load cells identifiers in Figures 7.1-4 are organized in a special order to show the simulation prediction, and the two measurements for the load cells that, in an ideal case, would be symmetric. The figures show also the error bar for each load cell which equals to 1% FSO (4.5 KN). This error is due to the sensor characteristics and not the measuring process.

Figures 7.1 and 7.3 compare the cover side load cells readings from test and simulation at clamping loads of 1575 and 2452 respectively. The comparison leads to several conclusions:

- Good symmetry is achieved between load cells on both sides of die symmetry plane, except for load cells C8 & C13. Perfect symmetry was not expected due to slightly different tie bars loading and due to different flatness values on platen, die and fixture plate surfaces. As mentioned before the load cells are very sensitive to the flatness of the surfaces they are in contact with.
- At the low clamping load, load cell C13 reads zero, while at the high clamping load it reads very low value compared to C8. The reason is that, at no-load condition, it is noticeable that this load cell is not in perfect contact with the plate surface. And since it was impossible to reach this load cell after assembly we could not shim it.

- A good match is noticed between the simulation model predictions and the experimental measurements, except at load cells C1, C15, C5, C10 and C13. The maximum loading in the simulation was at load cell C1 – and by symmetry C15- while in the experiment, the maximum loading was at C5 and C10. This is because of difference in clamping conditions between the model and the actual set up in the machine. In the machine, standard die clamps are used to clamp the die to the platen. These clamps are not modeled in the simulation due to the complexity of modeling them and the expected computational problems due to their contact with the die and the platen. In the simulation model clamps are replaced by tying the two surfaces in contact with each other. Considering modeling the die clamps explicitly may be a good idea in continuing this research.

Figures 7.2 and 7.3 compare the ejector side load cells readings from test and simulation at clamping loads of 1575 and 2452 respectively. The comparison leads to several conclusions:

- A good symmetry is achieved between load cells on both sides of die symmetry plane except for load cells (E6 & E16). The reasons for non-symmetry were discussed earlier.
- A good match is noticed between the simulation model predictions and the experimental measurements, except at load cells E8 and E9. As shown in the figures, the maximum predicted load –by simulation- was at load cell E1 and by symmetry E11. The experiment results showed the same thing. Also the minimum predicted load –by simulation – was at load cells E8, E9 and E10. Again the experiment results showed the same trend.
- A significant difference is noticed between the simulation and the test at load cells E8 and E9. The model under predicts the load at these two load cells. This is possibly due to the effect of the ejection mechanism on the ejector die. The ejector mechanism may carry some load from the ejector die back plate and transfer it to the middle load cells. The ejector mechanism was not included in the simulation model.

Figures 7.5 & 7.6 illustrate the %error between the simulation predictions and the load cells readings for clamping load 2452 KN at cover and ejector side respectively. Where:

$$\%error = 100 * \left| \frac{\text{Predicted load} - \text{measured load}}{\text{measured load}} \right|$$

Load Cell ID #	1575 KN Clamping Load		2452 KN Clamping Load	
	Average Load (KN)	Standard Deviation (KN)	Average Load (KN)	Standard Deviation (KN)
C1	72	0.07	141	0.51
C2	89	0.07	144	0.52
C3	112	0.09	174	0.72
C4	130	0.08	203	0.25
C5	205	0.05	269	0.32
C6	35	0.02	61	0.39
C7	33	0.05	51	0.13
C8	96	0.16	122	1.21
C9	68	0.04	103	0.65
C10	191	0.10	254	3.65
C11	19	0.03	43	0.20
C12	31	0.03	48	0.17
C13	0	0.00	8	0.04
C14	64	0.04	99	0.22
C15	98	0.11	170	0.37
C16	102	0.11	155	1.17
C17	110	0.06	171	0.60
C18	99	0.05	169	0.24
E1	146	0.24	229	2.10
E2	122	0.08	184	2.40
E3	78	0.04	132	0.15
E4	117	0.03	114	1.31
E5	126	0.06	178	0.93
E6	126	0.06	178	0.93
E7	97	0.10	154	0.42
E8	44	0.04	67	0.80
E9	36	0.02	50	0.45
E10	14	0.03	33	0.09
E11	160	0.46	243	2.61
E12	87	0.07	146	0.11
E13	103	0.05	153	1.20
E14	102	0.07	150	0.72
E15	85	0.06	130	0.14
E16	62	0.22	102	0.27
E17	87	0.06	145	0.27

Table 7.1: Load cells readings at 1575 KN and 2452 KN clamping loads

Load Cell ID #	Expected Load (KN) at 1575 KN clamp	Expected Load (KN) at 2452 KN clamp	Load Cell ID #	Expected Load (KN) at 1575 KN clamp	Expected Load (KN) at 2452 KN clamp
C1	143	222	E1	156	243
C2	99	154	E2	136	212
C3	114	178	E3	104	162
C4	113	176	E4	88	137
C5	80	125	E5	83	129
C6	57	89	E6	89	139
C7	46	71	E7	85	132
C8	52	81	E8	22	34
C9	60	94	E9	8	12
C10	80	125	E10	10	16
C11	57	89	E11	156	243
C12	46	71	E12	136	212
C13	52	81	E13	104	162
C14	60	94	E14	88	137
C15	143	222	E15	83	129
C16	99	154	E16	89	139
C17	114	178	E17	85	132
C18	113	176			

Table 7.2: Expected load cells loading from the simulation model

Figure 7.1: Comparison between measured and predicted load cells loading in cover side for clamping load 1575 KN

Figure 7.2: Comparison between measured and predicted load cells loading in ejector side for clamping load 1575 KN

Figure 7.3: Comparison between measured and predicted load cells loading in cover side for clamping load 2452 KN

Figure 7.4: Comparison between measured and predicted load cells loading in ejector side for clamping load 2452 KN

Figure 7.5: % Error for the cover side load cells at 2452 KN clamping load

Figure 7.6: % Error for the ejector side load cells at 2452 KN clamping load

7.1.2 Tie bar strain gages

Table 7.3 shows the average of the readings and the standard deviation for the measured strains in each tie bar. The Table 6.6.also shows the simulation model predicted values for the longitudinal strains in the tie bars.

	Clamping Load 1562 KN			Clamping Load 2418 KN		
	Average	Standard	Simulation	Average	Standard	Simulation
	measured strain (μ strain)	Deviation (μ strain)	Prediction (μ strain)	measured strain (μ strain)	Deviation (μ strain)	Prediction (μ strain)
TB1	261	0.42	233	386	0.57	361
TB2	259	0.39	275	404	0.57	426
TB3	244	0.17	233	367	0.82	361
TB4	265	0.37	275	409	1.25	426

Table 7.3: Strains at tie bars from strain gages and simulation model.

Figures 7.7 & 7.8 show comparisons between the simulation predictions and experimental measurements for tie bars strains at loads 1575 KN and 2452 KN respectively. The figures show very good symmetry between the tie bars on both sides of the symmetry plane. A very good match is also noticed between the simulation predictions and the experiment measurements.

Figure 7.7: Comparison between measured and predicted longitudinal strains in tie bars at 1575 KN clamping load

Figure 7.8: Comparison between measured and predicted longitudinal strains in tie bars at 2452 KN clamping load

7.1.3 Dies strain gages

Tables 7.4 & 7.5 show the dies strain gages readings and simulation model predictions respectively. Figures 7.9, 7.10 and 7.11 show a comparison between the predicted and measured strains in the dies for X, Y and Z directions.

Strain gage location	Direction	Measured strain (μ strain)	Strain gage location	Direction	Measured strain (μ strain)
CD1	Y	58	ED1	Y	11
	Z	-175		Z	-128
CD2	Y	46	ED2	X	97
	Z	-147		Z	-20
CD3	X	-93	ED3	Y	27
	Z	38		Z	-119
CD4	Y	51			
	Z	-47			
CD5	Y	44			
	Z	N/A			

Table 7.4: Measured strains in the cover and ejector dies at 2452 KN clamping load

Strain gage location	Direction	Simulation predicted strain (μ strain)	Strain gage location	Direction	Simulation predicted strain (μ strain)
CD1	Y	64	ED1	Y	32
	Z	-179		Z	-184
CD2	Y	67	ED2	X	100
	Z	-204		Z	-22
CD3	X	-45	ED3	Y	32
	Z	14		Z	-184
CD4	Y	67			
	Z	-204			
CD5	Y	64			
	Z	-179			

Table 7.5: Simulation predicted strains in the cover and ejector dies at 2452 KN clamping load

Figure 7.9: Comparison between measured and predicted strains in dies at 2452 KN clamping load (X-direction)

Figure 7.10: Comparison between measured and predicted strains in dies at 2452 KN clamping load (Y-direction)

Figure 7.11: Comparison between measured and predicted strains in dies at 2452 KN clamping load (Z-direction)

Figures 7.9-10 show a generally good match between the simulation predictions and the strain gages readings. All the strain gages show the same strain patterns, although the values are different. Figure 7.9 shows a very good match between the simulation predictions and the strain gages readings at the ejector side. The reading at the cover side, though showing the same strain pattern, has a significant different value from the simulation prediction. For the Y-direction strains, given by Figure 7.10 all the strain gages show a good match except ED1. For the Z-direction strains, given by Figure 7.11 a good match is noticed except for strain gages CD3 and CD4. CD5 strain gage did not work.

Comparing the results from tie bars and from dies we can conclude that the strain gages readings on tie bars match the simulation predictions much better than the die strain gage readings. Several reasons may contribute to this fact. The first expected reason is the high surface finish of tie bars compared to the rough surface finish of the dies. The surface finish is a very important factor for the strain gage mounting and hence its accurate reading. A second reason is that, the model assumes that the die casting machine is symmetric around one plane and hence only half of the machine is modeled. Although the tie bars strains proves that the tie bars loading is symmetric, the dies may not be that symmetric and hence different readings are expected. The third reason is that, in the simulation model the dies are very much affected by the boundary conditions than the tie bars. The type of contact definition between the dies and platens can cause artificial effects on the strains in the dies.

7.2 Load Condition II - Casting

At this load condition, actual casting process took place. 67 castings were produced. The clamping load was 2500 KN, and the intended cycle time was given in Table 6.2. Since pouring, spraying and extraction were done manually, the actual time was different than 40 seconds. However, most of the cycle times ranged between 40 and 60 seconds.

7.2.1 Load Cells

Figures 7.12 and 7.13 show the readings from four load cells at each side during one cycle. The figures also show the simulation prediction for the same load cells. The selected load cells are a very good representation for the other load cells

(a)

(b)

Figure 7.12: Load pattern at load cells C3, C17, C7 and C12 during one cycle

(b)

Figure 7.13: Load pattern at load cells E1, E11, E5 and E15 during one cycle

Figure 7.12 demonstrates that the load cells readings at the cover side decreases after intensification. This applies to all load cells on the cover side. The summation of the readings of the load cells at the cover side decreased by 10% after intensification. On the other hand the simulation results showed different patterns. The load cells behind the cavity predicted higher load after intensification, while the load cells far from the cavity predicted lower load. The summation of the load cells predicted loads by the simulation is constant and it is not affected by the intensification pressure.

The pattern in the ejector side was different than on the cover side, as shown in Figure 7.13. The load cells readings reflected no significant change in the readings after adding

intensification. Again the simulation predictions were different. The upper load cells of the die predicted higher load after intensification, while the lower load cells predicted lower load. In the simulation and the experiment the summation of the load cells reading at the ejector side did not change after intensification.

The main reason for the differences between the simulation and experimental measurements patterns, after intensification, is the stiffness of the machine. In the model the machine is stiffer than what it is actually. Figure 7.14 shows a schematic drawing and a free body diagram for the machine. In the machine the tie bars and the toggle system are attached to a third platen at the back of the machine. The deformation of this platen due to the intensification pressure relaxes the load slightly on the cover platen and stretches the tie bars. In the simulation the rear platen is not modeled. The tie bars and toggle system are completely restrained at their ends in the space and hence no deformation occurs at their ends.

Another difference between the machine and the model is the support frame behind the cover platen. Three support bars transform the load from the cover platen to the support frame, Figure 7.14. The support frame also carries the reaction force from the piston during intensification. The support frame and the three support bars are not modeled.

Figure 7.14: A schematic drawing and a free body diagram of the machine

7.2.2 Tie bars strain Gages

Figure 7.15 shows the strains measured in the tie bars and the strains predicted by simulation for one casting. The figure 6.shows a very good match between the simulation model predictions and the strain gage readings. The figure also shows that the strain in the tie bars increases after intensification. The reason for this attitude is explained in the previous section.

Figure 7.15: Strain pattern at tie bars TB1&TB3 (a) and TB2&TB4 (b) during one cycle

7.2.3 Dies Strain Gages

Figures 7.16, 7.17 and 7.18 show a comparison between the measured strains in the cover and ejector dies and the simulation predictions for X, Y and Z directions respectively. The figures show that there is good match at some locations, while some other locations show completely different patterns between simulation and experiments. The reasons for the different readings in the die strain gages were discussed previously. Another factor for noise results predicted by the simulation is the interaction between the stress analysis and the temperature distribution. The large differences that can be seen in Figure 7.16 and 7.17 occurred in the simulation due to the effect of temperature. This looks like a numerical error rather than an actual change in the strain.

Figure 7.16: Comparison between measured and predicted strains in dies at load condition II (X-direction)

Figure 7.17: Comparison between measured and predicted strains in dies at load condition II (Y-direction)

Figure 7.18: Comparison between measured and predicted strains in dies at load condition II (Z-direction)

7.2.4 Thermocouples

As mentioned earlier in this chapter, four thermocouples were inserted in both dies to measure the temperature inside the die steel. The locations of the thermocouples are given in Figure 6.7. Due to problems with the cement used to fix the thermocouples in the fixtures, they did not work as intended. Only thermocouple T3 worked correctly. Figure 7.19 shows the readings of thermocouple T3 for 14 cycles. Figure 7.20 shows the simulation predictions of temperatures for the same thermocouple. A very good match in the pattern is shown between the two figures although some differences in the values are noticed. Simulation predictions are 30°C lower than the thermocouple measurement after the same number of cycles. Many reasons may contribute to this difference. The first expected reason is the furnace temperature. The furnace used in the die casting facility is old and controlling its temperature is a challenge. It is very likely that the pouring temperature was higher than what was assumed in the simulation. The second reason is the manual control of the pouring and idle time periods. These two time periods are constant in the simulation, while in the actual casting they change significantly. The third reason is related to the model where the heat transfer coefficients between casting/inserts, inserts/dies and inserts/air may be different from the actual values. Considering all these sources of variability, the simulation predictions are considered very consistent with the experimental measurements.

Figure 7.19 Measured Temperature Profile

Figure 7.20 Simulated Temperature Profile

APPENDIX A - Specifications

A.1 Load cells specifications

The load cells were purchased from OMEGA-DYNE. The model selected is LC307-100K. Load cells properties are given in Table A.1. Figure B.1 shows the load cell. The load cells calibration was performed by the manufacturer, OMEGA-DYNE, and a calibration sheet was provided for each load cell.

Property	Value
Range	0-100,000 lbs
Safe overload	150,000 lbs
Ultimate overload	300,000 lbs
Linearity	1% FSO
Zero balance	2% FSO
Operating temperature range	-54 to +121 °C
Compensated temperature range	+15 to +71°C
Excitation	5 V dc

Table A.1: Load cells specifications

Figure A.1: Load cell, shape and dimensions. Dimensions are in inches (mm)

A.2 Thermocouple Specifications

The thermocouples specifications are given in Table A.2.

Property	Value
Manufactures	OMEGA
Model	XC-20-K-24
Type	K
Size	20 gage
Range	(-185) - (+1250)°C

Table A.2: Thermocouples specifications

Table A.3 shows the specifications of the used cement used to fix the thermocouple in its fixture.

Property	Value
Manufactures	OMEGA
Coefficient of thermal expansion	4.6E-6
Maximum service temperature	843 °C
Thermal conductivity	1.15 W/mK
Electrical conductivity	Insulator

Table A.3: Cement properties

A.3 Strain gages specifications

Property	Value
Manufactures	OMEGA
Model	SG-7/350-LY11
Maximum strain	30000 μ strain
Hysteresis	negligible
Service temperature (static)	-30-250 °C
Service temperature (Dynamic)	-30-300°C

Table A.4: Strain gages specifications

A.4 Signal Conditioning Device Specifications

Seven signal conditioning devices (Model # SCXI-1520, from National Instruments) were used to collect the data interchangeably from the load cells and strain gages. Table A.5 shows the specifications of the SCXI-1520. One signal conditioning device (Model # SCXI-1102C, from National Instruments) was used to collect the data from the thermocouples. The specifications of this device are given in Table A.6. A DAQ device (Model # PXI-6052E, from National Instruments) was used for collecting data from the signal conditioning devices and performing the A/D conversion. Table A.7 gives the specifications of the PXI-6052E.

Property	Value
Manufactures	National Instruments
Model	SCXI-1520
Analog Input Channels	8
Data Sampling	Simultaneous
Programmable Excitation	1-10 V
Programmable Gain	1-1000
Programmable Lowpass filter	10HZ-10KH

Table A.5 Specifications of SCXI-1520

Property	Value
Manufactures	National Instruments
Model	SCXI-1102-C
Analog Input Channels	32
Data Sampling	Multiplexed
Preset Lowpass filter	10KH

Table A.6 Specifications of SCXI-1102-C

Property	Value
Manufactures	National Instruments
Model	PXI-6052E
Analog Input Channels	16
Data Sampling	Multiplexed
Sampling Rate	333KS/s
Resolution	16-bit
Input Range	± 0.05 to ± 10 V

Table A.6 Specifications of PXI-6052E

Appendix B Fixtures Drawings

Figure B.2: Cover intermediate plate

Figure B.3: Cover platen plate

Figure B.4: Ejector die plate

Figure B.5: Ejector intermediate plate

Figure B.6: Ejector platen plate

Figure B.7: Thermocouple fixture. Internal part (a), external part (b)

Appendix C Load Cell Setup

Figure C.1: Load cells at the cover side

Figure C.2: Load cells at the ejector side

Figure C.3: Cover die setting

Figure C.4: Ejector die setting

Figure C.5: Both dies are installed on the machine

Figure C.6: Sensors wires are protected in temperature resistant looms

Appendix D LABVIEW Program

Figure D-1: Labview program main window

Figure D-2: Labview program diagram window

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