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**DIE CASTING DIE DEFLECTIONS:
PREDICTION AND ATTENUATION**

Final Report for the Period 7/1/95 through 9/30/97

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**R. A. Miller
H. Ahuett-Garza
A. K. Choudhury
S. Dedhia**

May 1998

Work Performed Under Contract No. DE-FC07-95ID13361

**For
U.S. Department of Energy
Assistant Secretary for
Energy Efficiency and Renewable Energy
Washington, DC**

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**By
The Ohio State University
Columbus, OH 43210**

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ABSTRACT

The following report summarizes two years of research intended to develop methods to model and predict the deflection patterns in die casting dies. No comprehensive analysis of this type had previously been completed.

The die casting process is complex and involves numerous mechanical and thermal phenomena that effect the mechanical behavior of the die. A critical activity in this work was sorting out and evaluating the relative contributions of the various mechanisms to die deflections. This evaluation was accomplished through a series of simple engineering analyses based primarily on the order of magnitude of the influence of each load considered on die deflections.

A modeling approach incorporating commercially available finite element analysis software was developed and tested. The model evolved by testing simple models against more comprehensive models and against the limited experimental data that is available. The development of the modeling approach lead to consideration of the die casting machine in more detail than was originally anticipated. The machine is critical and cannot be ignored. A simplified model described as a spring/platen model was developed to account for the machine platens, tie bars, and toggles. The characteristics of this model are described and predictions based on this model are compared against full machine models and measured deflections of machine platens.

Several computational experiments, organized using design of experiment techniques, were performed to pull out the key die and machine parameters. Results of these experiments are presented and they show that the deflections as characterized by the maximum separation at the parting plane of the die casting die is very sensitive to the uniformity and rigidity of the die supports. The key variables appear to be the machine platen thickness, cover die thickness, and the position of the die on the platen. Several other variables are shown to influence parting plane separation, but the three mentioned above appear at this point to be the most important factors.

Details of the modeling approach and the various case studies are provided in the report and in several publications that have resulted from the work.

ACKNOWLEDGEMENTS

This work was supported and monitored by the members of the North American Die Casting Association's Computer Modeling Task Group chaired by Bill Walkington of Walkington Engineering. The contributions of the members of the committee were invaluable and the work could not have been completed without this help.

Special thanks are due Walter Smith of DCD Technologies for sharing expertise and helping to focus some of the issues. Thanks are also due Warren Bishenden, Exco Engineering; Charles Ramsey, Matrix Technologies; Dick Smith, Chrysler; Greg Prince, Walker Die Casting; and Danny Swartz, Delaware Machinery; and Bob Hegel and Bill Damian of Prince Machine Co. All of these individuals and companies provided expertise, experience and data used for validation of the results of this work. Professor Ted Allen of the Industrial, Welding and Systems Engineering Department at OSU assisted with experimental designs and response surface analysis for the die parameter and machine parameter computational experiments. Ted's help was instrumental in unraveling the relationships among the many design parameters.

The help of the committee, and particularly the individuals mentioned above, has contributed to the quality of the work. Any errors or inaccuracies, however, are the responsibility of the authors.

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1. SUMMARY AND INTRODUCTION

Die casting dies deflect during a casting operation and these deflections affect the operation of the casting process and the quality of the part. The goal of this work was to model and predict die casting die deflections. The engineering problem is that of determining the elastic deflections of a body under periodic thermal and mechanical loads.

The main objective of this work was to develop modeling procedures to predict die casting die deflections under processing conditions. This is a complex objective because the range of phenomena in the die casting process is very large. As a consequence engineering judgement must be used to develop practical, tractable models.

The first task in this work was to sort out the interactions among the many different loads, and to determine the relative significance of each loading phenomenon. This is important since it provides an engineering basis for ignoring certain phenomena while considering others. It is computationally infeasible to consider all loads in full detail.

The second major issue addressed is the determination of the degree to which the characteristics of the die casting machine that mechanically supports the die must be explicitly modeled in order to account for die deflections. Several models are considered ranging from a simple rigid support model, to a fairly complete model of the machine including the tie bars and platens. It is shown that the simpler rigid-support models dramatically underestimate the deflections and fail to account for the pattern of separation at the parting plane.

A modeling approach was developed using ABAQUS, a commercial FEM system. The required parameters and boundary conditions have been defined. Simulation results have been compared favorably with operational experience and field data. Anyone with access to a FEM system can follow the approach developed in this work and carry out die simulations.

Several case studies were completed to provide data that can be used for design purposes. These case studies also provide insight into the general die deflection phenomenon. In addition to providing specific data, the results help to provide a better general understanding of the ways that dies respond and some of the things that can be done to minimize the problems.

Chapter 2 provides a detailed order of magnitude analysis of the various phenomena that create mechanical and thermal loads in die casting dies. This chapter provides the basis for the modeling approach. Chapter 2 is based on the Ph.D. dissertation of Horacio Ahuett-Garza (Ahuett-Garza 1996; Miller and Ahuett-Garza 1997). This chapter may be skipped by those interested only in application of the results.

Chapter 3 outlines the basic finite element modeling approach and provides an illustration of the use of the approach using a simple flat plate die. Those interested only in the application of the modeling techniques developed may wish to start with chapter 3.

Chapter 4 provides several case studies that serve to partially validate the work and to establish the level of modeling detail required. Slide blowback (slide movement) data was available for a slide, which had a horn pin design. This data was used for model

validation and for a design case study. The results show that the modeling approach works quite well. Also in chapter 4, a series of models representing different methods of accounting for the support provided by the die casting machine and the support it provides to the die. These results are compared and used to show that the flexibility of the machine is very significant and must be included in the overall model. This work also serves to evaluate the merits of several measures of die performance including maximum separation at the parting plane, average contact pressure, and maximum contact pressure. This work is based in large part on the masters thesis of Sanjay Dedhia (Dedhia 1997; Miller, Choudhury et al. 1997).

A model that accounts for the elasticity of the machine was constructed using spring elements to model the platen support behind the dies. The model was validated with measurements of platen deflection provided by Prince Machine Co. The model and the results are discussed in chapter 5. This work is based on the masters thesis of Aswin Choudhury (Choudhury 1997; Miller, Choudhury et al. 1997). The American Metalcasting Consortium supported a portion of this work but it is included in this report to provide a comprehensive summary in one place to maximize the benefit to the die casting industry.

Die casting die design is an art, which depends on the experience of the die designer. Die designers determine the size and shape of the individual components based on thumb rules or personal experience. Formal treatment of the structural design of dies in the literature is not very extensive. In chapter 6 a few small first steps are made to analyze the sensitivity of the die deflection to several die design variables. The studies include a sensitivity (design optimization) study of horn pin lock designs, response surface analysis of key variables in an open-close die, and a response surface analysis of the effect of selected die casting machine variables on die deflections. This chapter summarizes the best indications to date of the significance of die and machine variables with respect to die deflections and parting plane separation.

The results of these studies are not complete and represent only the first steps in establishing solid engineering data. Continuing work will refine the recommendations and lead to more complete data.

The modeling approach however has been shown to be credible and can be utilized by practitioners interested in improving die designs. Based on the results of this work, practitioners have information that can be used in setting up other simulations and data that can be put to use immediately.

2. CHARACTERIZING THE FORCES IN DIE CASTING

2.1 *Introduction*¹

In this section, the loads that act on the structure of a die in a typical die casting operation are introduced and mathematical descriptions of the phenomena in question are developed. Simplifications and assumptions to generate boundary conditions for finite element modeling are made.

The first, and perhaps most important, obstacle in predicting the behavior of the die is the existence of widely varying time scales and size scales. The time scales are due respectively to:

- fill time on the order of milliseconds,
- solidification time, on the order of seconds,
- casting cycle on the order of seconds to minutes
- casting process operation time, on the order of hours

The size scales are:

- size of the features in the cavity, millimeters,
- size of the thermal layer subject to transients within each cycle, centimeters
- size of the region where temperatures fluctuate significantly in the long run - same order of magnitude as the size of the die, meters.

In essence, the maximum deflections in the die are a function of the stiffness of the die and its supporting structure. This stiffness depends upon three factors:

- fundamental design of the die
- clamping force
- die temperature

In general, a die deflects and changes shape in response to forces due to clamping, pressure resulting from fill and solidification, and growth due to the accumulation of heat. The die changes shape near the cavity in response to heat released by solidification. The next paragraphs provide a description of the loads and the time and dimension scales in which they have an effect. Figure 2.1 illustrates many of the loads that act upon the structure of a die casting die.

2.2 *Locking Action*

At the instant of closure, the machine presses the ejector and cover sides of the die together. The locking force is exerted by a toggle mechanism. Figure 2.2 presents a macroscopic model of the clamping action of the machine upon the die. Once the toggle

¹ This chapter is based primarily on the Ph.D. dissertation of Horacio Ahuett-Garza (Ahuett-Garza 1996).

Figure 2.1 Depiction of loads present in die casting

is engaged, the locking action pre-loads the die in preparation for the application of pressure generated by the injection stage. The locking force of the machine determines the magnitude of this pre-stress. This macroscopic model of the locking action of a die casting machine predicts a behavior which is analogous to the behavior of a bolted joint and it is useful for the purpose of computing the shutoff force (F_s) needed to hold a die together. Clearly, dies will be closed as long as F_s is positive. This model however cannot predict the spatial nature of P_s , the shutoff pressure, at the parting surface. P_s determines the likelihood of losing shutoff area, and determines the ability of the die to be free of flashing.

This simple model illustrates that the fact that the compression of the toggles results in stretching of the tie bars. This combination of the tie bars and the toggles places a preload on the dies via the platens. The preload establishes the initial shutoff force. As shown in figure 2.2b, as pressure is applied to the cavity, the load on the machine linkage (and the tie bars) increases and the shutoff force decreases until the shutoff force is lost at the point that the cavity load becomes too large. The magnitudes of these loads will change as the die grows due to heat accumulation from the casting over multiple cycles.

(a)

(b)

Figure 2.2

a) Depiction of the clamping force

b) Characterization of the relationships between loads

A simulation to predict the magnitude of the deflections of the die needs to account for the loads generated by the locking action of the machine, particularly if the subject of study is the parting plane of the die. At first sight, an order of magnitude analysis shows that the thermal stresses in a die are given by:

$$\sigma \cong E \cdot \alpha \cdot \Delta T$$

In particular, ΔT represents the rise in temperature of the region around the cavity that defines the thermal interaction of the die and casting. Under typical conditions:

$$E = 2.1 \times 10^{11} \text{ (Pa) (Modulus of Elasticity)}$$

$$\alpha = 15 \times 10^{-6} \text{ (}^\circ\text{C}^{-1}\text{) (Coefficient of Thermal Expansion)}$$

$$\Delta T = 250 \text{ (}^\circ\text{C)}$$

Therefore,

$$\sigma = 7875 \times 10^5 \cong 1 \times 10^9 \text{ Pa}$$

By comparison and based on typical die/machine arrangement settings², stresses generated at the parting surface are of the order of $2 \text{ E}+06 \text{ Pa}$. This load is almost 2 orders of magnitude smaller.

A more careful look will show that this is true only if one neglects the true interaction between both loads. For the thermally induced loads to reach such high values, the die must be constrained from growing in the direction of the application of the locking force. This will necessarily be accompanied by an increase in the locking force. These interactions can be effectively accounted for with a finite element model.

In summary, it is necessary to account for the effects of the clamping force when simulating die casting die deflections. This is particularly important if the goal is to establish the behavior of the die at the parting plane.

2.3 Filling Stage

During fill, molten metal is injected into a die cavity in an extremely short time. In aluminum die casting for example, liquid metal may be ladled at 650°C and forced to fill the cavity in less than 0.05 sec. In general, these conditions result in flow regimes within the die cavity that surpass the onset of turbulence. From the perspective of our analysis, the filling process is important only to the degree to which a die is loaded by filling phenomena. In other words, the mechanisms of fill need to be described with a resolution that is significant from the point of view of their effect upon the magnitude of die deflections. From this same point of view, the characteristics that define molten metal flow are the hydrodynamic pressure and momentum transfer generated during the filling stage, plus the heat released by the metal as it solidifies.

² Based on clamping force and die size data published on the Bühler Cold Chamber Die Casting Machine Catalog.

The definition of the flow characteristics is not a straightforward task. Traditionally, researchers have implemented solutions of the Navier Stokes equations, coupled with the mass conservation and energy equations to describe the filling mechanisms (see Appendix A). Except for a few cases, no analytical solution exists for these equations. In the case of die casting, the presence of jet or atomized flow requires that the boundaries of the fluid front in the vicinity of obstacles be computed as part of the solution, thus complicating the problem immensely.

Implementation of numerical solutions for this system in arbitrary domains is extremely complicated unless simplifying assumptions are made. The use of these equations implicitly carries the assumption that molten metal behaves like a Newtonian fluid, in spite of the fact that in a die casting operation molten metal carries a considerable amount of solid and entrapped gas. An argument may be made that the system is in fact a multi-phase slurry. Nevertheless, these equations seem to provide reasonable results when predicting the characteristics of flow for gravity casting operations. The validity of using this formulation when predicting the characteristics of high pressure die casting additional investigation. Fortunately, this level of detail is not required for this work.

Characterization of the loads during fill consists of estimating the relative contributions of each load to the overall deflections of the die. Under certain conditions, a scale analysis similar to the one presented by (Eckert 1989) can be helpful. The objective is to establish the order of magnitude of the loads. To differentiate between the contributions of two different loads of the same order of magnitude, other techniques must be used. For example, the system of equations (A.1 through A.5, appendix A) may be solved under certain conditions using a numerical simulation based on a finite difference scheme.

Given the difficulties in solving the equations necessary to predict the filling regime and the pressures, temperatures and forces exerted upon the die, two questions must be addressed:

1. Without solving the fill equations them explicitly, what kind of statements can be made about the nature of the loads upon the structure of the die? and
2. How accurate does the description of the loads need to be in order to allow one to model die deflection with some degree of confidence?

These questions will be addressed throughout the rest of this document.

2.3.1 Heat released during fill

The simplest way to analyze the filling stage is by assuming that this process takes place under isothermal conditions. In essence, Equation A.4 and those that follow are eliminated from the problem formulation. The flow regime can then be established by solving the Navier Stokes Equations (A.1). This is still a formidable task given the conditions found in a typical die casting operation.

(Eckert 1989) tried to develop a generic description of the mechanisms that take place during the filling stage. With a scale analysis, he concluded that the ratio of t_s , the time scale for solidification, to t_c , the time scale for conduction within the die, is given by:

$$\frac{t_s}{t_c} = \frac{H_{Al}}{c_{p-Die} \Delta T_{Al-Die}}$$

Typical values of these parameters in a typical die casting operation are:

$$H_{Al} = 390,000 \text{ J/kg}$$

$$c_p = 590 \text{ J/kg-}^\circ\text{C}$$

$$\Delta T = 350^\circ\text{C}$$

where ΔT represents the typical temperature difference between die and molten metal. From this equation:

$$\frac{t_s}{t_c} = 1.9$$

which means that the time scale for conduction is of about the same order of magnitude as the time scale for solidification. This fact illustrates the point that the heat transfer characteristics of the die do not hinder the solidification process. Die casting belongs to the class of processes whose solidification time is governed by the characteristics of the interface between the molten metal and the die. Referring to Figure 2.3, the heat released by the casting is absorbed by the die through the finite conductance imposed by the interface. Assuming the molten metal is exactly at its liquidus temperature (i.e. no superheat) and based on Equation A.5, the total heat transferred into the die per unit area is given by:

$$\rho H \frac{\partial S}{\partial t} = h(T_{Al} - T_{Die}) = h\Delta T$$

where S is the length of solidified metal measured from the casting-die interface, H is the latent heat of the solidifying material and h is the thermal resistance of the interface. Solving for S :

$$S = \frac{h\Delta T}{\rho H} \int_0^{t_s} dt$$

Assuming that the time scale for solidification is defined as the time in which the solidification front reaches the middle of the casting (refer to Figure 2.3), then:

$$S = \ell$$

where ℓ is half the thickness of the part³

³ Given that this is a scale analysis, this parameter may also refer to the average thickness of the part, or the smallest thickness of the part, when there are significant differences between the smallest and largest thickness.

$$t_s = \frac{\rho H}{h \Delta T} \ell \dots \dots \dots (2.1)$$

the time scale for fill is given by

$$t_f = \frac{L}{V}$$

and finally

$$\frac{t_s}{t_f} = \frac{\rho H \ell V}{L h \Delta T} \dots \dots \dots (2.2)$$

Figure 2.3 Schematic model for definition of a simple solidification front

Typical values for die casting are:

$$\begin{aligned} \rho_{Al} &= 2570 \text{ kg/m}^3 \\ h &= 40,000 \text{ W/m}^2\text{-}^\circ\text{C} \\ H &= 390000 \text{ J/kg} \\ \Delta T &= 350 \text{ }^\circ\text{C} \end{aligned}$$

or

$$\frac{t_s}{t_f} \cong \ell \cdot \frac{70}{t_f}$$

It would be expected that heat released during fill would not be significant as long as $\frac{t_s}{t_f} \geq 10$, which requires $\frac{\ell}{t_f} \geq \frac{1}{7}$ (where ℓ is in meters and t_f is in seconds). In this case, the fill time is at least an order of magnitude faster than the time required for solidification. Heat released during fill would not play a role in the magnitude of the deflections of the die under these conditions. In other words, from the point of view of the magnitudes of deflections, the analysis can assume that the molten metal starts at a uniform temperature within the cavity. As a corollary, fill should proceed without thermally induced problems (such as cold shuts) whenever $\frac{t_s}{t_f} \geq 2$. This approximation is

based with the knowledge that under typical conditions molten aluminum will carry a significant fraction of solid during fill (40%-60%). The assumption is that solidification time is about twice the time it takes to fill the cavity.

Clearly, the last statement stretches the validity of Eq. 2.1, given all the assumption made in arriving to this equation. Nevertheless, this number is consistent with the observations reported by (Chen 1991) reproduced in Figure 2.4. This figure illustrates how a majority of castings in a typical plant are produced under conditions that the previous formulas indicate will not have fill related problems. By the same token, the figure also shows that most castings will fill in a regime in which heat released is not insignificant.

It is important to establish that the degree of significance depends on the effect that is being analyzed. As it will be shown, even in the case in which large thermal gradients exist within the molten metal once the cavity is full, thermal fields and deflections are affected only in a small scale.

An issue arises in those cases in which there is a significant amount of heat released during fill (i.e. when $\frac{t_s}{t_f} \cong 2$): how are the deflection patterns affected by this deviation from the instant fill condition? A significant amount of heat released during fill may affect die deflections in two ways:

1. within the time frame of a single cycle, the temperature fields, and possibly the deflection patterns in regions close to the cavity may differ significantly from those found under "instant fill" conditions.
2. within the time frame of the whole casting operation, the quasi-steady state-temperature fields in the bulk of the die may be affected, resulting in a different stiffness of the die.

The existence of the different time and size scales is depicted by the previous two statements. Given the variation in part geometry, die design and process conditions found in die casting, it is difficult to make broad statements about the effect of heat released during fill.

Figure 2.4 Scale analysis of heat released during fill. Data from (Chen 1991)
Reprinted with permission of NADCA.

Barone and Caulk (Barone and Caulk 1993) have studied this problem from the point of view of its effect upon the thermal patterns in the die structure. Consistent with their previous work, an exact description of filling or solidification patterns within the cavity is not developed. In their original work, a computer model was developed to estimate die temperatures during quasi steady state without the need to simulate multiple cycles. That work assumed instant fill. More recently, they have added to their model the capability of computing the liquid metal velocities during fill. These velocities are averaged across the cavity thickness. In their paper, there is no description of how the flow field is computed. However, it is clear that this is not done in a rigorous manner (i.e. solving the Navier-Stokes equations). Instead, it appears that the casting is developed into a surface, from which a flow field is established based on a constant average velocity, which in turn may be computed by the ratio of distance to the farthest point from the gate to fill time. Temperatures are assumed constant throughout the thickness of the casting. A full thermal analysis consists of two steady conduction analyses, with a transient heat conduction analysis for thermal patterns during fill in between. This analysis provides the correction from the instant fill solution that is used to establish thermal fields in the first heat transfer analysis.

Barone and Caulk also present a two dimensional case to illustrate the effect of heat released during fill. Their results indicate that only under very extreme circumstances do the thermal fields in the die deviate significantly from those obtained when instant fill is assumed. Their work however reports only average temperatures in the die, without any

reference as to how average die deflections are affected. Furthermore, their simulations do not seem to include the effects of coolant spray upon the die.

2.3.2 The effects of heat released during fill

A simulation with a geometry similar to the one used by Barone and Caulk was set up to establish the effect of heat released during fill, while taking into account the effect of coolant spray. The geometry and casting conditions are shown in Figure 2.5. The transient problem being solved is essentially:

$$\rho c_p \frac{\partial T_{n,\alpha}}{\partial t} = k \left(\frac{\partial^2 T_{n,\alpha}}{\partial x^2} + \frac{\partial^2 T_{n,\alpha}}{\partial y^2} \right)$$

with convection boundary conditions:

$$-k \frac{\partial T_{n,\alpha}}{\partial x_i} = h(T_\infty - T_{x_i})$$

where:

n = cycle number

α = 1,2 for die and casting respectively

i = coefficient that identifies the coordinate of the normal direction at interface

Because a change of phase is involved, the specific heat (c) is given by:

$$c_p = \begin{cases} c_o & T_{n,2} > T_{\text{liquidus}} \\ c_o + \frac{H}{(T_{\text{liquidus}} - T_{\text{solidus}})} & T_{\text{liquidus}} > T_{n,2} > T_{\text{solidus}} \\ c_o & T_{\text{solidus}} > T_{n,2} \end{cases}$$

The two problems are coupled by the term:

$$-k_1 \frac{\partial T_{n,1}}{\partial x_i} = h(T_{n,1} - T_{n,2})$$

at the die-cavity interface. For each cycle n , $T_{n,1}(\text{initial})$ equals $T_{n-1,1}(\text{final})$ from the previous cycle. In addition, $T_{n,2}(\text{initial})$ is prescribed as 630 °C for the instant fill case.

In the non-instant fill case there was no explicit flow simulation to compute thermal patterns in the casting or die. Instead, an unbalanced initial temperature distribution $T_{n,2}(\text{initial})$ was used to account for heat released during fill. This unbalance is computed by assuming that molten metal loses heat uniformly as it travels from the entrance to the end

of the cavity. Consequently, a volume pack of liquid metal at the entrance to the cavity accumulates heat from all the volume packs that have passed through it. In this case, it was assumed that the first volume package to enter the cavity loses the equivalent of 50% of its latent heat by the time it reaches the end of the cavity. Therefore, the volume packages at the end of the cavity will have less heat by a margin of 50% latent heat, while those packages that are close to the gate will have more heat by the same margin. Volumes at the symmetry plane are unchanged with respect to the instant fill conditions.

While this heat loss may seem extreme, it is known that under normal conditions there is a significant amount of solid (40% +) carried within the molten metal. Although in practice 50% or more of latent heat may be lost from the molten metal from the furnace to the cavity, it would be extremely difficult and unlikely to reach the imbalance in the temperature assumed in this exercise. At any given point the temperature is given by:

$$T_{n,2} = 630 \pm \frac{F}{c_p}$$

where F is the fraction of heat to be added or removed from the corresponding volume. The range of temperature between the coldest and the hottest region of the casting exceeded 250 °C. To put this gradient in perspective, a simulation of the three-dimensional version of this geometry (a plate) with MAGMA shows a maximum gradient of about 80 °C between the coldest and the warmest region (the biscuit). The fill time in this case is more than twice as long as would be considered typical for the application.

Returning to the 2 dimensional model, for each case, instant fill and non instant fill, 15 cycles were simulated. ABAQUS was used to carry out the simulations. Assumptions regarding the casting conditions are also shown in Figure 2.5.

Figures 2.6 and 2.7 illustrate the rate at which quasi-steady (periodic) conditions are reached for each case. These figures show that at the end of 15 cycles, the maximum temperature change by any node from the conditions at the end of the 14th cycle is less than 5 °C.

Figure 2.8 shows the temperature fields and deflection patterns for instant fill conditions at the end of 15 cycles. Figure 2.9 does the same for non-instant fill. As would be expected, the instant fill case shows an almost perfectly symmetrical thermal field when compared to the non-instant fill case (slight deviations are found primarily due to the fact that the mesh is not perfectly symmetric). The deflection fields are also slightly different. Figure 2.10 illustrates the differences between the two cases. The nodal temperatures were obtained by subtracting the results of the instant fill case from those of the non-instant fill case. The range between the coldest and the warmest spot in this plot (Figure 2.10 (a)) is more than 100 °C, or roughly a third of the maximum steady state temperature of the bulk of the die. This occurs in a region that spans approximately one third of the length of the parting plane. The resulting difference in the deflection fields (Figure 2.10 (b)) reaches 8 E-05 m (approximately 0.003 in). This value is within the range of the manufacturing tolerance for the surface in question. While this dimension is not

Notes:

Cooling lines' cross sections are .01 x .01
 Casting thickness = 0.005
 All dimensions in meters

Simulation Conditions:

Casting Material:	Al 380
Liquidus Temperature	580 °C
Solidus Temperature	520 °C
Latent Heat	390 KJ/Kg
Specific Heat:	963 J/Kg °K
Heat removed by spray:	625 E+03 J/m ² /sec for 3 seconds (20% of heat)
Initial Temperatures:	Instant fill: 630 °C
	Non instant fill: TA = 830°C
	(50% of latent heat, more heat content than instant fill)
	TB = 561°C
	(50% of latent heat, less heat content than instant fill)

Figure 2.5 Testing the effect of instant fill vs. non instant fill conditions.
 Description of geometry and conditions.

insignificant with respect to the change in dimensions due to the rise in bulk temperature of the die, it is apparent that its effect lies within a range that belongs in a different degree of detail. Two interesting facts are also evident from this analysis. First, as seen in Figures 2.8 and 2.9, the shape of the parting plane at operating conditions differs by up to 6×10^{-4} m (0.024 in) from manufacturing dimensions. Clearly, the shutoff pressure of this arrangement is much different than would be expected if the parting plane was flat. In general, die designers do not account for this factor in their design. Furthermore, manufacturing tolerances are much tighter than this value. This suggests that the performance of shutoff surfaces may be improved by compensating for this growth rather than by improving manufacturing or assembly tolerances.

Second, a true quasi-steady state, in which periodic conditions are reached, requires a very large number of cycles. Figures 2.6 and 2.7 show the projected number of cycles to reach periodic conditions. In both cases it takes about 125 cycles for maximum temperature differences between consecutive cycles to drop under 1°C . In their own exercise, Barone and Caulk report that it took up to 300 cycles to reach quasi-steady state. The implication is that true periodic conditions would be extremely difficult to achieve in practice. Consequently, dies should be designed to operate within a characteristic range of temperatures.

Geometric arrangements and process conditions found in die casting vary widely. Consequently, deviations in the thermal fields in actual cases from what this exercise has shown can be expected. Significant heat released during fill will occur only when a part is very large compared to its thickness. Based on Eqs. A.6 and 2.1, conditions defined in this exercise correspond to:

$$t_s \approx 2 \cdot t_f$$

which results in:

$$t_f \approx \frac{\rho H \ell}{2h\Delta T} \approx 35 \cdot \ell$$

where ℓ is in meters.

Despite the limitations of this exercise, two conclusions may be drawn:

- Under long fill times, heat released during fill is significant.
- The magnitude of its effects is nevertheless a small fraction of the overall heat effects on clamping of the die. The magnitude of the effects of heat loss during fill is on the order of other sources of noise.

Maximum temperature difference with respect to
previous cycle
Case: Instant fill

a)

Maximum temperature difference with respect to
previous cycle
Case: Instant fill, Projected

b)

Figure 2.6. a) Maximum temperature difference for any node between consecutive cycles, for instant fill conditions. b) Projected number of cycles to reach periodic conditions.

Maximum temperature difference with respect to
previous cycle
Case: No instant fill

a)

Maximum temperature difference with respect to
previous cycle
Case: No instant fill, Projected

b)

Figure 2.7. a) Maximum temperature difference for any node between consecutive cycles, for non instant fills conditions. b) Projected number of cycles to reach periodic conditions.

Figure 2.8. Deflection patterns at end of 15 casting cycles. Instant fill conditions.

Figure 2.9. Deflection patterns at end of 15 casting cycles. Non - instant fill conditions.

Figure 2.10. Deflection patterns illustrating the difference between instant fill and non instant fill conditions.

2.3.3 Momentum transfer during fill

During fill, molten metal impinges on different regions of the cavity, losing speed and changing direction. Velocities and concentration of fluid reach their largest magnitudes at the gate, where the cross section of the travel path is the smallest. Given their geometry and location with respect to the gate, slides are prime examples of components exposed to momentum generated forces. In fact, it is difficult to visualize a component that could be more susceptible to impact induced deflections than a slide tip placed directly in front of a gate.

In general, metal flowing into the cavity will not form a continuous stream. Air entrapped within the flowing metal, changes in the plunger acceleration during fill, and non uniform transitions in the cross section of the channels that the fluid follows are some of the factors that cause a discontinuous stream. Consequently, the slide tip will be intermittently loaded during a time so short that the slide may be unable to deflect in response to the short bursts of force induced by the flowing metal. The most critical case would be a situation in which a continuous stream impinges upon the tip. This case is illustrated in Figure 2.11. To estimate the force upon the tip, a volume can be defined in the vicinity of the gate, in which a momentum balance is performed, neglecting all external and drag forces:

Schematic of Slide Tip/Gate Arrangement Modeling the tip as a beam

b) Magnitudes of deflection, based on beam models.

$$I = \frac{bh^3}{12}, P=70 \text{ MPa}, F=10,000 \text{ N}.$$

Model	Stiffness :: K (N/m)	Deflection m :: (in)
1. 	$\frac{48EI}{L^3}$ (1344 E +06)	7.44 E -06 (0.0003)
2. 	$\frac{3EI}{L^3}$ (84 E +06)	119 E -06 (0.0047)
3. 	17.6 EI (493 E +06)	20.3 E -06 (0.0008)
4. 	$\frac{AE}{L}$ (8400 E +06) F=PA	333 E -06 (0.013)

Figure 2.11. Analysis of momentum related deflections. a) Model of tip / gate / fluid stream. b) Table describing magnitude of deflections for different slide tip arrangements.

$$\frac{dM}{dt} = - \int_{A_1} (\rho v) v \cdot dA - \int_{A_2} (\rho v) v \cdot dA - \int_{A_1} p dA - \int_{A_2} p dA \dots\dots (2.3)$$

where:

M = Momentum

p = pressure

v = fluid velocity

A_{1,2} = Area at inlet, outlet of control volume

Assuming that the tip is close enough for the continuous stream to maintain the cross sectional area defined at the gate, and assuming steady state, Eq. A.6 for the given geometry and coordinate system reduces to:

$$\frac{dM_x}{dt} = \rho(v_1)^2 A_1 \cos \alpha - \sum_{i=1}^2 (v_{2,i})^2 A_1 \cos \alpha + p_1 A_1 \cos \alpha - p_2 A_2 \cos \alpha$$

Assuming further that there is perfect venting in the cavity, the order of magnitude of the force due to momentum can be defined as:

$$F = \rho(v_1)^2 A_1 \cos \alpha$$

as a function of the mass of the casting (m), fill time (t_f) and area of the gate (A_g), and for small angles (a):

$$F = \rho \left(\frac{m}{\rho A_1 t_f} \right)^2 A_1 = \frac{m^2}{\rho A_1 t_f^2}$$

It can be observed that the magnitude of the force is very sensitive to the ratio of mass vs. fill time. Typical values for large castings, such as an aluminum (ρ = 2570 kg/m³) transmission case are:

$$m = 20 \text{ kg}$$

$$A_1 = 0.001 \text{ m}^2$$

$$t_f = 0.2 \text{ sec}$$

from which F is approximately 4000 N. Typical values in small castings are:

$$\begin{aligned}m &= 2 \text{ kg} \\A_1 &= 0.0001 \text{ m}^2 \\t_f &= 0.05 \text{ sec}\end{aligned}$$

which results in $F = 6225 \text{ N}$. Based on these two cases, the force due to momentum can be assumed to be in the order of 10,000 N. To determine the magnitude of the deflection of the slide tip, the stiffness of the slide/tip arrangement must be determined.

The order of magnitude of the stiffness can be estimated based on beam theory. In general, slide tips that are placed in front of a gate are provided with an extra support at their tip. Tips that are not impinged upon directly by the stream coming from the gate do not necessarily have an extra support. As a first approximation, a cantilever beam can be used as model. The deflection at the tip of the cantilever is given by:

$$\partial = \frac{FL^3}{3EI} \dots\dots\dots (2.4)$$

from which the stiffness can be defined as:

$$K = \frac{3EI}{L^3}$$

Assuming a rectangular cross section of height (h) and base (b), Formula 2.4 becomes:

$$\partial = \frac{F}{E} \cdot \frac{4L^3}{bh^3}$$

For typical applications, L is of the order of 1 m, while b and h are of the order of 0.2 m at the smaller end of the tip. Then, for a steel tip ($E = 210 \text{ E} + 09 \text{ Pa}$):

$$\partial = \frac{10 \text{ E} + 03 \cdot 4 \cdot 1^4}{210 \text{ E} + 09 \cdot 0.2^4} = 120 \text{ E} - 06 \text{ m} = 0.0047 \text{ in}$$

This value is of approximately the same order of magnitude as the manufacturing tolerances for the details in the tip, and is well below the expected distortion due to the relaxation of thermal stresses once the part cools down to ambient temperature after ejection. Figure 2.11 shows different beam models and their corresponding stiffness for a particular case. Case 1 perhaps provides the most accurate description of the order of magnitude of the slide tip deflections under momentum loads. Interestingly, for the given geometry the contraction induced by a typical cavity pressure (assuming it was applied to

the full area at the end of a cantilevered tip) would be higher than the largest deflection caused by impact.

Based on these results it can be concluded that, in general, momentum induced deflections are relatively small. Only in poor designs would deflections due to impact of the flowing metal become significant. On the other hand, momentum forces may exceed the weight of some of the components upon which they act. While this force may not induce significant strains within the tip, the tip may move as a rigid body if there is clearance between the slide and the cover die.

2.3.4 Hydrodynamic pressure

Of all the loads in a die casting application that have been studied in this work, hydrodynamic pressure is the one that has received the least attention in the literature. Estimating the nature of this load presents exactly the same complications as establishing the amount of heat released during fill. Strictly speaking, the same equations need to be solved. Given the arbitrary shapes and conditions found in die casting, specific cases will present different behavior. In fact, from the point of view of the die, the unbalanced load caused by a hydrodynamic pressure may not be more severe than the natural unbalance that arises from the combination of an arbitrary geometry and the fluid path that fills it. An estimation of the relative contributions of each one of these types of loads associated with the filling flow cannot be accomplished without looking at a variety of different cases. However, and without being rigorous, a few comments may be made about the characteristics and effects of hydrodynamic pressures during fill.

Contrary to the characteristic viscosity dominated flow found in gravity casting or injection molding, the filling regimen in die casting is an inertia dominated process. This is particularly true in the cavity region. In the runner region fluid velocities are significantly lower than at the gate or within the cavity. Ignoring the effect that the cavity geometry has upon the magnitude and direction of the force as seen by the die structure, the maximum pressure drop during fill would be caused by changes in part thickness. The largest such change occurs at the entrance to the cavity.

Runners are designed to blend into the gates smoothly, maintaining a constant cross section. Molten metal accelerates as it moves from the biscuit region towards the gate. However, once metal flows past the gate, the area for flow increases dramatically. At this point the metal is traveling at such a high speed that under typical conditions it will break apart and bounce off the cavity surfaces. In practice, gates often present a discontinuity in the path of flow. It is conceivable then that the gate defines the boundary between two regions where pressures differ significantly from each other. It is very likely that pressure drops within other regions of the die would be smaller than at this region. In the limit, the maximum imbalance would result in a load of similar magnitude as that produced by momentum induced forces in front of the gate.

In establishing the importance of the hydrodynamic pressure, there are other factors to consider. As has been discussed, filling occurs in a very short time. Variations in the pressure during fill will take place in times that are shorter than the actual fill time. The different die components may not have enough time to respond to these variations. Furthermore, if a given component were able to move in response to hydrodynamic pressure, it is conceivable that it would be able to return to its equilibrium position once

the unbalance is removed. It will be shown that a hydrostatic pressure distribution seems to explain the deflections reported in field data. Under these conditions, most components would be free of unbalance once the cavity is full, and therefore should be able to return to their original conditions.

Based on the available evidence, it is difficult to justify the effort needed to prepare a more accurate description of the deflections caused by a hydrodynamic pressure. A full scale study could be justified if the study of other phenomena (for example, formation of porosity) were included as part of the objectives of such an analysis.

2.3.5 Pressure surge at end of fill

For many years, casters have witnessed deflections that seem to coincide with the end of fill. Casters associate this deflection with a pressure surge that can be sometimes measured in the hydraulic system of the machine. This pressure spike, as it is commonly called in industry, has been known to burst pressure gauges before the magnitude of the spike is actually measured.

Figure 2.12 shows a schematic of the hydraulic actuation of the plunger in a die casting machine. In essence, the plunger is a piston that moves under the application of pressure through the hydraulic fluid. During fill, the hydraulic system is capable of generating plunger velocities of 10 m/s or more. Given the short duration of the actual injection cycle, large accelerations and decelerations are usually experienced by the plunger. Pressure transients are thus expected within the hydraulic system and, more importantly, within the die cavity.

One source of dynamic loads is the kinetic energy of the moving masses within the system. In the more basic die casting machine and casting operations, only back pressures exerted by molten metal upon plunger tips as cavities fill up slow plungers down. Clearly, all masses that are in motion (i.e. plunger and hydraulic fluid) must come to rest at the end of fill (or shortly thereafter, given that a small oscillation within the plunger-hydraulic fluid-molten metal may be expected). Eventually, all dynamic loads must be dissipated by the structure of the die and machine.

The other source of dynamic loads is precisely the casting machine's hydraulic system. Sudden rises in pressure within the hydraulic system at the end of cavity fill have been documented and reported on numerous occasions (Everett 1972; Kawaguchi and Tanaka 1977; Everwin 1987; Herman 1988; Hatamura, Hirahige et al. 1989; Mickowski and Teufert 1993). This pressure spike is a result of the sudden interruption of the flow of oil as the plunger comes to rest, and is of a similar nature to the water hammer phenomena in pipes. As early as 1972, systems for the suppression of this spike in the hydraulic system of the die casting machine were designed (Everett 1972).

Of particular relevance to this research is the fact that, to this date, there is considerable confusion about the role that pressure surge within the machine's hydraulic system plays in regards to the quality of a casting operation. Many casters are under the impression that flashing is the result of this pressure spike. Several experimental studies (Kawaguchi and Tanaka 1977; Everwin 1987; Mickowski and Teufert 1993) have confirmed the existence of a pressure spike within the cavity of the die just at about the instant that the cavity is completely full. However, these studies have also shown that the pressure rise

within the cavity does not coincide with pressure spikes in the hydraulic system. Investigators associated with these studies argue that while the presence of shock absorbers (water hammer suppressers) reduce the amplitude and period of the hydraulic vibration after plunger arrest, their effect upon the pressure surge within the cavity is minimal (Kawaguchi and Tanaka 1977). The implication is that, from the perspective of die deflections, the largest load upon a die at the end of fill is caused by a sudden dissipation of kinetic energy as the plunger is brought to rest. This would support the observation that the existence of large pressure spikes in a machine's hydraulics does not necessarily result in large die deflections (Bishenden 1995). Today, many if not all of the die casting machine builders provide their machines with some type of plunger deceleration device that reduces the plunger speeds right before they come to a complete stop.

Referring to Figure 2.13, a considerable back pressure will not exist until the runner system begins to fill. This rise may not occur at a constant rate during fill time and, given the characteristics of the filling flow, it is not uniform (spatially) throughout the cavity. The die does not feel the full extent of pressure until the cavity has been filled completely. Conversely, the plunger is not subject to the maximum back pressure until this last instant.

(Barton 1967) argued that the kinetic energy associated with the moving masses, or at least a significant portion of it, is dissipated in the form of heat. According to his calculations, the temperature rise in the molten metal produced by this energy may exceed 50 °C. There is no solid experimental evidence to support his claims. There are two other ways in which this energy may be dissipated:

1. In the form of sharply localized, traveling disturbances, i.e. elastic waves, that are eventually damped out within the die and machine.
2. By inducing deflections within the structure of the die. Once all energy is dissipated, the die would return to an equilibrium state. The time it would take to reach this state would depend upon the damping characteristics of the system.

Case (1) would not produce a significant (macroscopic) deflection in the die. Other issues may arise though (for example, unwanted vibrations and noise). Case (2) clearly falls within the scope of this research. The key is being able to estimate the magnitude of the maximum pressure within the cavity.

In his doctoral dissertation, (Sachs 1952) analyzed the die casting process from the perspective of the hydraulic characteristics of the die casting machine and die. His work consisted of a one dimensional analysis of the characteristics (pressure and velocity) within the system. Clearly, this analysis is capable of predicting these characteristics only at a macroscopic level. Assuming fluids are non-elastic, Bernoulli's equation for unsteady flow is:

$$f(\xi) \frac{\partial c}{\partial t} \int \frac{\partial \xi}{f(\xi)} + \frac{c^2}{2} + z + \frac{p}{g} = F(t)$$

Figure 2.12 Schematic diagram of hydraulic actuation of die casting machine.

where:

ξ = coordinate along conduit

$f(\xi)$ = area of conduit (variable)

c = velocity of fluid along conduit

z = height of fluid particle with respect to reference

p = pressure of fluid particle

g = acceleration of gravity

$F(t)$ = integrating function, independent of location

A balance of forces at the time the cavity begins fill, when pressure upon the plunger starts to build up is:

$$A_H p_H - (A_H - A_B) p_B - A_O p_O - R_s = M \frac{dv}{dt}$$

Referring to Figure 2.12, the subscript "H" refers to the locations behind the plunger, "b" refers to the are of the plunger extension, "o" refers to the plunger tip region, "M" and "v" are the mass and velocity of the plunger respectively. While it may be argued that this equation holds throughout the injection cycle, the reality is that a significant p_o will be encountered only when the runner system begins to fill up. This factor becomes even

more pronounced once metal begins to flow into the cavity. The same may be said about R_s .

Any of the given pressures can be presented as a function of the accumulator pressure (P). For example, according to Sachs, pressure behind the plunger (p_B) has the form:

$$p_H = P - \rho \frac{v^2}{2} f(r, x) - \rho \frac{dv}{dt} g(r, x)$$

where $f(r, x)$ and $g(r, x)$ are functions that account for friction (r) within the system as a function of plunger displacement (x).

For the most part, these equations are not solvable analytically, and simplifications must be made. In his work, Sachs eliminated a variety of factors that were supposedly unimportant, and determined that the relationship between the accumulator pressure (P) and the air cavity pressure (p) was of the form:

$$K_1 P - K_2 p = K_3 \frac{dv}{dt} + K_4 v^2$$

where K_n ($n=1,2,3,4$) are constants that account for pressure losses in the system. Field data was available to Sachs, and he concluded after computing a numerical solution of the previous equations that the dependency upon plunger displacement proved to be negligible, and so terms affected by this variable were dropped from the equation. Once the K_n constants are known, pressures at different location within the system can be evaluated.

Clearly, these equations include a multiplicity of parameters that characterize a particular machine and die. While some of these factors may be available (especially those associated with commercial components such as valves and pipes), others are more difficult to find, specifically those associated with the cavity. For these equations to have any value as tools for predicting cavity pressures, significant experimental data needs to be generated regarding flow resistance factors within the cavity.

Sachs showed that this approach provides first order approximations to the actual pressures and velocities, which may vary from results generated by more rigorous methods (e.g. based on a wave formulation) by less than 3%. The limitation of this approach is that pressure spikes/water hammer mechanisms cannot be accounted for with this formulation.

Assuming that the pressure surge within the cavity is caused mainly by the dissipation of the kinetic energy of the moving masses, an alternate approach for computing the magnitude of the spike within the cavity can be developed around the actual profile of the plunger velocity. A case in point is presented in Figure 2.13. This chart presents the history of plunger velocity and pressure at the tip vs. plunger position. This trace was recorded during the injection of a transmission casing. The capability of the casting machine to decelerate the plunger was not used in this case.

Obviously, the hydraulic drive of the die casting machine is designed to overcome all sources of friction and back pressure (p_B) within the piston cylinder in such a way that the plunger can be rapidly accelerated to injection speeds. As has been mentioned, the cavity

pressure will eventually provide the force that brings the plunger down to full stop. Assuming that molten metal behaves inelastically, the magnitude of the pressure surge within the cavity can be computed as the force at the plunger end divided by the plunger area. Based on the assumption of non-elasticity of the fluid and given the ability of the molten metal to take the shape of the cavity, the pressure spike is perceived as an increase in the hydrostatic pressure within the cavity. Referring to the free body diagram in Figure 2.14, a summation of forces on the plunger gives:

$$\sum F_x = m\ddot{x}$$

and

$$F_H - F_C = m\ddot{x}$$

Based on the arguments presented before, the change in kinetic energy equals the work that the cavity pressure exerts upon the plunger in order to stop it:

$$\begin{aligned}\text{Spike energy} = \Delta K.E. &= \int F_{\text{spike}} dx \\ \frac{mv^2}{2} &= \int (F_C - F_H) dx \\ &= - \int m\ddot{x} dx\end{aligned}$$

and consequently

$$F_{\text{spike}} = - m\ddot{x} \dots\dots\dots (2.5)$$

and

$$P_{\text{c-spike}} = \frac{F_{\text{spike}}}{A_{\text{plunger}}} \dots\dots\dots (2.6)$$

The plot in Figure 2.13 displays only velocity vs. position. In order to extract acceleration from this chart, one can use the following form:

$$\frac{dx}{dt} = \frac{dx}{dx} \cdot \frac{dx}{dt}$$

or as a first order approximation

$$\ddot{x}_i = (\text{slope at point } i) \cdot (\text{velocity at point } i)$$

where (i) refers to a specific point along the abscissa.

The effects of force generated by the resistance to flow within the cavity have not been included in this analysis. Consequently, spike forces computed by this method will be an upper bound or conservative value.

The force needed to bring the plunger down to a complete stop equals its rate of change of momentum. The pressure spike is then computed as this force divided by the cross sectional area of the plunger. Figure 2.15 shows the approximated values of velocity and cavity pressure as a function of time. These plots were based on Figure 2.14 and were developed using Eqs. 2.4 and 2.5. In this particular case there is not a clear spike. Instead, the plunger seems to come to rest rather smoothly in a very short period of time after a series of small spikes. It is known that plungers vibrate as cavities fill. The smaller spikes are an indication of these vibrations, and may be caused by sudden increases in back pressure upon the plunger tip as the molten metal flows through the gate. In the case presented in Figures 2.13 and 2.15, the magnitudes of these spikes may be accentuated by the fact that line segments were used to fit plunger velocity data. Consequently there is no continuity in the slope between adjacent line segments (i.e. there is no continuity in the acceleration).

From the perspective of the definition of a load in a finite element model, the question that needs to be answered is which load is larger: injection pressure *plus* the pressure surge within the cavity at the end of fill or intensification pressure? (As has been explained before, other studies have shown that pressure surges in the hydraulic system of the machine do not affect the cavity pressure). In the majority of the traces that have been observed, plungers tend to decelerate rather smoothly. Under these conditions, intensification pressures that are at least 35% higher than injection pressures will cause the largest deflections.

Abrupt changes in the slope of the plunger velocity trace indicate the presence of discontinuities in the acceleration profile. These changes may indicate the presence of large pressure spikes. It would be very expensive to try to eliminate the effects of these spikes by increasing the size of the die. As a consequence, pressure spikes must be dealt with in a different manner (such as with the use of plunger deceleration devices).

Clearly, it would be desirable to have a tool that can, analytically and a priori, estimate the magnitude of the pressure spike at the end of fill. At first look, Sach's approach seems to provide an answer. In reality, values of resistance coefficients that characterize the behavior of the machine hydraulics would have to be determined experimentally. Furthermore, resistance coefficients for the cavity geometry would have to be developed.

Because of the lack of data in the literature, a study would have to be conducted to first identify the factors that affect the velocity profile, and second to determine equations based on these factors for the prediction of velocity profiles in die that have not been built yet. More important than the velocity profile would be the estimation of the maximum plunger deceleration as the cavity fills up, since this parameter will determine the maximum in pressure cavity at the end of fill.

Figure 2.13 Field data: velocity and pressure vs. plunger position. Data provided by Exco, Engineering (Newmarket, Ontario, Canada).

Figure 2.14. Forces acting on plunger.

Figure 2.15 Estimated plunger velocity, force and cavity pressure

2.4 Dynamic Response of Die and Platen

The importance of the pressure spike depends upon the magnitude of the deflection it causes. With the objective of describing how a typical system die / platen responds to a load of characteristics that are similar to what would be expected from the pressure surge at the end of fill, two independent structures were simulated.

Figure 2.16 illustrates the geometry and dimensions for each model. IDEAS was used to model the response of the structures to a pressure spike, using the Lanczos method to extract the natural frequencies.

Figures 2.17 and 2.18 present a description of the excitation and the response of the structure recorded in 3 points. The excitation is basically a force that is applied during τ seconds, and requires $\tau/2$ seconds to reach its peak value. For each structure, τ and the damping coefficient were varied, while the peak force on the structure was kept constant for all cases. Peak deflection was recorded for each case. τ for Figures 2.19 and 2.20 is 0.001 sec. These figures also show the response under a static load (that is, a load that is applied very slowly) of the same peak magnitude.

The response to the static load can also be interpreted as the steady state response of the structure. The results of this analysis are summarized in Figures 2.21 and 2.22. The plots show the ratio of the peak magnitude of deflection under dynamic conditions to the deflection under static load as a function of the spike duration. In essence, these plots show how dynamic a certain excitation is. For example, Figure 2.21 shows that for a spike that lasts 0.001 seconds and assuming the structure is completely undamped, the deflection caused by this dynamic load is 40 % higher ($\delta_{\max}/\delta_{\text{sta}} = 1.4$) than the deflection that the same load would produce if it were applied very slowly.

In general, simulation results show that the die / platen system responds to the load in a quasi-static manner when the time to peak value is approximately 0.005 sec. or longer (note that the time to peak under our simulation conditions is half the spike duration). It is thus clear that typical intensification pressures will not result in significant dynamic effects (intensification pressure rise takes more than 50 milliseconds). For lightly damped structures (damping factors less than 0.5 of critical damping), pressure may have important effects when the time to peak lies between 0.4 and 2.5 milliseconds. Loads associated with a pressure spike at the end of fill are more likely to lie in this region, and therefore would produce important dynamic effects. Clearly, a structure that is capable of responding to such short burst of energy would also be able to recover very rapidly once the load is removed. From the charts spikes that last between 0.008 and 0.005 seconds would have potentially the most important dynamic effects (i.e. flashing may be worse under such conditions).

Figure 2.16 Ejector and cover die/platen arrangement used for dynamic analysis

Figure 2.17 Cover side: excitation and response under two different conditions.

Figure 2.18 Ejector side: excitation and response under two different conditions.

Figure 2.19 Shock response curve for cover side.

Figure 2.20 Shock response curve for ejector side.

The above curves provide a first order approximation to estimate dynamic effects of pressure spikes in die casting. From an experimental study whose goal was to measure adverse effects associated with the pressure surge, (Everwin 1987) reported pressure spikes that took about 0.0015 seconds to reach their peak values. The in-cavity pressure spikes that he reports are very similar in shape to the pressure spikes used in this section. This would translate into a spike whose duration is approximately 0.003 seconds. The shock response curves report an overshoot (conservative-no damping) anywhere between 20 and 35% for this spike duration. This would be consistent with what is commonly observed in industry, which is illustrated by the fact that die casting machine builders (Damian 1996) usually recommend that clamping forces be 20% higher than a normal analysis based on injection pressures alone would report.

2.5 Summary of the Loads in Die Casting

The conclusions reached in each section of this chapter can be summarized to provide a picture of the characteristics of the loads and die behavior in a die casting operation:

- From the perspective of their effects upon the magnitude of die deflections, impact forces from the liquid metal and hydrodynamic loads are negligible under typical casting conditions. Only poorly designed structures or uncharacteristic filling conditions would make these loads important.
- Injection pressure may induce a dynamic response in the die structure under very specific circumstances. For example, a typical die/800 ton machine arrangement will likely suffer dynamic effects when pressure build up within the cavity takes less than 0.005 sec. Pressure spikes within the cavity in such cases will produce an overshoot when compared to the steady state response for the same magnitude of loads.
- Typical patterns of intensification pressure build up clearly result in quasi-static loads. In the majority of cases, maximum deflections are reached with intensification.
- Compared to instant fill conditions, heat released during fill may produce variations in the thermal fields of the casting and die that are not necessarily insignificant. However, the contributions are probably not more significant than other sources of noise (for example, manufacturing and assembly tolerances).

The analysis presented in this chapter reduces to the following recommendations for modeling die deflections:

- Intensification pressure and heat released from solidification have to be considered.
- The effects of heat released during fill can be ignored.
- Dynamic effects have to be addressed in a different manner than by increasing the stiffness of the die.
- Intensification pressure is more significant than momentum.

3. MODELING DIE CASTING DIE DEFLECTIONS

3.1 Introduction

The objectives in this chapter are:

- to present a methodology to simulate die deflections.
- to describe the issues related to preparing a reliable model
- to present alternatives to address these issues

Based on the analysis of a flat plate die, and with the help of field data, the behavior of a die casting die will be characterized. As will be shown, this characterization is consistent with what is commonly observed in the field. The analysis of slide blowback and the analysis of additional castings will be presented in later chapters.

3.2 Steps in Simulation

Figure 3.1 shows a flowchart for a typical simulation session. Preparation of a model involves the definition of geometry, mesh, boundary conditions, load history and simulation control parameters. The centerpiece of this chart is the numerical computations that result in the description of the state of stress/strain of the problem at hand. The numerical methods that are commonly applied to the analysis of uncoupled thermelasticity problems (Finite Element, Boundary Element) are relatively reliable tools that have been tested in other fields of study. It is the other steps in the chart that still present challenges for the application of such numerical methods for the analysis and simulation of die casting die deflections.

Throughout this research, ABAQUS (from Hibbit, Karlsson & Sorensen, Inc.), a commercial finite element analysis package, was used to carry out numerical computations. Most models were prepared in IDEAS (SDRC, Milstead, Ohio). In spite of the flexibility and power of these two systems, the quality of the simulation depends totally on the decisions made at the moment the model is prepared for simulation.

In the next sections, the issues that arise in preparing a model for simulation and the manner in which they can be addressed will be illustrated through a specific example.

3.3 A Specific Example: Geometry and Casting Conditions

Figure 3.2 presents the geometry of a die and flat plate casting. This geometry is a scaled up version of an experimental die available at the Ohio State University. The dimensions of the original die were modified to match an 800 ton die casting machine. Two versions of this die were simulated. In one, the insert on the ejector die was modeled explicitly. In the other, the ejector die was assumed to be a monolithic, solid, entity. That is, it was assumed that the cavity was machined directly into the ejector base. The two cases were designed to test the importance of explicitly modeling inserts.

Figure 3.1 Flowchart for typical simulation

Figure 3.2 Flat plate geometry (dimensions in inches)

Tables 3.1 and 3.2 illustrate the process conditions and characteristics of the casting cycle that were used in this exercise, respectively. Heat transfer coefficients were collected from the literature (Caulk 1990; Papai and Mobley 1991; Barone and Caulk 1993; Kock 1993).

Table 3.1: Assumed casting conditions

	Final Simulations (Full Die)
Casting Material	Al380
Melt Temperature	600 °C
Maximum Cavity Pressure	70 Mpa (10.000 psi)
Cycle Time Analyzed	36 sec
Clamping Force	7250 KN (815 tons)
Ambient temperature	20 °C
Heat transfer coefficient steel-air	20 J/°K
Heat transfer coefficient steel-steel	2000 J/°K
Heat transfer coefficient steel-cooling line water	10000 J/°K
Temperature of platens	80 °C
Water temperature	40 °C

Table 3.2: Assumed steps in a single casting cycle

Time (sec)	Step
0-1	dies are clamped
1-17	heat and pressure applied to cavity
17-32	die opens, part is knocked out
32-35	lubricant spray
35-36	dies clamped

A die deflection simulation consists of two steps.

1. A heat transfer analysis is conducted. In this step, only heat transfer boundary conditions are included in the model.
2. Temperature fields from step 1 are added to a stress analysis model, in which all mechanical loads and corresponding boundary conditions are included.

3.4 Model Geometry and its Effect on Mesh Size

Preparation of an FEM model starts with the creation of model geometry. As is the case with many CAD/CAE systems, IDEAS has the capability of semi-automatically generating a mesh of finite elements out of a volume. Nevertheless, complex geometry may cause meshing problems, such as unacceptable degrees of distortion in the finite elements, or extremely large mesh sizes. Accuracy of results is greatly affected by these two factors.

The most important mesh control parameter is the characteristic length of the element. In general, fine meshes (i.e. small elements) can reproduce part geometry better and are less distorted than coarse meshes. Consequently, simulation results are more accurate. On the other hand, fine meshes require more computational power. Therefore, a compromise must be made between accuracy of simulation results and the cost of generating them.

A technique to reduce meshing problems involves the elimination of small rounded edges in the model of the casting. In the case of the flat plate die, elimination of these features resulted in geometry whose smallest edges was approximately 0.01 m long. Consequently, the range of feature sizes was reduced by about an order of magnitude. Figure 3.3 (a) and (b) show the original casting and the modified version that was used for simulation purposes, while Figure 3.3 (c) illustrates the model for the die. Because of the symmetry of the problem, only half of the geometry was actually modeled.

It is important to understand the effect that the elimination of rounded edges can have upon simulation results. By replacing smooth features with sharp edges, maximum stresses will be overestimated in regions close to these geometric features. Deflection results, on the other hand, are less sensitive to slight modifications in the geometry. While strains may be overestimated, displacements will not be greatly affected as long as these strains remain localized.

When more complex geometry is being studied, and if results are sought at locations far enough from the cavity, complete features may be ignored. This technique will be illustrated by the exercise in slide blowback presented in a later chapter.

It should be pointed out that creating the geometric model and mesh is generally a time consuming procedure. If the goal is to have a tool that can be used at the design stage, simulation/analysis systems whose architecture is not open to accept geometry from CAD systems are seriously handicapped.

Figure 3.3 Wireframes of a) casting, b) modified geometry, and c) die

3.5 Load History and Boundary Conditions

3.5.1 Heat released during solidification

There are at least three possible ways to compute temperature fields in die casting dies. The obvious approach would be to run a solidification simulation with a specialized system (i.e. MAGMASoft⁴, ProCast⁵, EKK⁶, and others). Because most of these systems are not designed to address die stress analysis, temperature data must be exported from this system into a model for stress analysis based on another system (such as ABAQUS). The need to export data between two different systems requires that the architectures of both systems be open and compatible, or that an external tool that provides a suitable interface be available. The meshes used in each analysis must also match, or mesh interpolation software available. At present transferring data in any of these ways is not straightforward.

The simplest approach to calculate die temperatures consists of carrying out a heat transfer analysis in which the magnitude of the heat from solidification is prescribed. Numerous experimental studies (Hong, Backman et al. 1979; Chijiwa and Shirahige 1981; Lee, Nguyen et al. 1991; Papai and Mobley 1991) have shown that heat released by the casting into the die proceeds approximately at an exponentially decaying rate, irrespective of location. In his work, (Padiyar and Miller 1994) took advantage of this behavior to model the thermal load upon the die. His basic assumption was that heat escaping from the part flows in a direction perpendicular to the cavity surface only, and that the part is very thin.

In this model, the constraint is that the total heat released by the part be governed by the following equation:

$$H = C_l (T_m - T_l) + \text{Heat of Fusion} + C_s (T_s - T_e) \dots \dots \dots (3.1)$$

where:

H = Total heat released by the part

C_{l,s} = Specific heat for liquid, solid casting material

T_m = Melt temperature

T_l = Liquidus temperature

T_s = Solidus temperature

T_e = Temperature at time of ejection

⁴ MAGMA Foundry Technologies, Inc. Arlington Heights, Ill

⁵ UES Inc. Dayton Ohio

⁶ EKK, Inc., Walled Lake, Michigan

It is assumed that the rate of heat release may be approximated by an exponentially decaying function, where the initial rate is several times larger than the rate at the end of the thermal cycle. The shape of the heat release is given by:

$$Q = K \cdot 10^{mt} \dots \dots \dots (3.2)$$

where:

Q = heat rate

K = arbitrary constant

m = parameter that defines the rate of release

t = time of heat release

This distribution must meet the constraint (3.1), and therefore:

$$H = \int_{t_o}^{t_i} K 10^{mt} dt \dots \dots \dots (3.3)$$

where:

t_o = time when heat release starts

t_i = time at which heat from solidification is stopped

As an initial approximation, one can assume:

$$\frac{Q(t_o)}{Q(t_i)} = 100 \dots \dots \dots (3.4)$$

Given a total heat release (H) for a given region, constants K and m in equation (3.2) can be computed based on the constraints imposed by equations (3.3) and (3.4). In essence, the accuracy of the approximation depends upon the values used for K and m . The trick is to match the part temperature at ejection (T_e) after a time ($t_i - t_o$).

This model may also be used to represent the heat extracted by the lubrication spray. It has been reported that, in general, the heat extracted by lubricant spraying ranges from 15 to 30 % of the heat delivered by the solidification process. The previous equations (3.1 through 3.4) apply once again, with the following considerations:

- H_s = Heat removed by spray
- $H_s = 0.15H$,
- t_o = time at which spray starts
- t_i = time at which spray is stopped

The major problem that prevents general application of this approach is the fact that the values of constants K and m are not the same for all regions of the cavity. This is due

mainly to the intricate shape of typical castings, which results in regions of varying volumes. Consequently regions such as runners and biscuits will necessarily have different heat release rates than the product.

3.5.2 Heat flux applied to flat plate die

With the tools available, the most practical method to define the thermal history in the die involves running a simulation with MAGMA (or similar program) and obtaining temperature data at selected locations.

Based on the temperature results, the total amount of heat released by the part can be estimated and values can be assigned for K and m in different regions. These values are used in a heat transfer simulation with ABAQUS. The thermal patterns predicted by ABAQUS can then be compared with the MAGMA results and the values of K and m can be modified if significant differences exist. This process is repeated until temperature fields in both simulations are similar. Figure 3.4 summarizes this procedure.

In all, only a couple of iterations (i.e. ABAQUS simulations) were needed to compute the values of K and m that resulted in thermal fields that matched approximately those reported by MAGMA. (Examples of this process with another casting are shown in Appendix B.) Table 3.3 presents the values used for the selected regions. With these values, thermal patterns within the die were found to differ by less than 70 °C on the cavity surface, and less than 30 °C in the bulk of the die, or about 10% of the temperature values reported by MAGMA at the given locations.

Contrary to what is done in the case of the heat given up by the casting, the heat removed from the cavity surface during lubricant spraying was modeled as a negative heat flux and applied on the free element faces in the finite element model. In our flat plate exercise, a value of 15 % was used. It was assumed that lubricant was applied for 3 seconds (i.e., $t_i - t_o = 3$).

Table 3.3 Thermal zones and respective K and m values

Thermal zone	During Solidification		Coolant Spray	
	K (W/m ²)	m (s ⁻¹)	K (W/m ²)	m (s ⁻¹)
Overflow	2.019 E+06	-2/15	-3.162 E+06	-0.5927
Fan	2.538 E+06	"	-3.978 E+06	"
Plate	2.168 E+06	"	-3.396 E+06	"
Runner	3.225 E+06	"	-5.099 E+06	"
Biscuit	3.332 E+06	"	-7.771 E+06	"

Figure 3.4 Flowchart to determine constants K , m

3.5.3 Other heat transfer boundary conditions

Accounting for convection/conduction between the different interfaces completed the heat transfer model for the flat plate die. It was assumed that convective heat transfer takes place at the cooling channels and the exterior of the die during the die holding time, lubricant spraying, and die open time. When the die is closed, the parting surfaces are in contact and their interface is represented by a finite conductance in the heat transfer model. When the die opens up, these surfaces are exposed to the ambient, and therefore a convective heat transfer boundary condition is added. Table 3.1. includes the values used for the heat transfer coefficients, as suggested by (Barone and Caulk 1993), and the assumed boundary temperatures.

3.5.4 Cavity pressure and clamping force

As a first approximation, clamping forces are treated as if they correspond to a uniformly distributed pressure at the back of the dies. Clearly, the accuracy of this assumption depends on factors such as the geometry and deflection of the ejector platen. The assumption basically treats the platens and machine as completely rigid.

The analysis uses intensification pressure as the maximum load that will be applied within the cavity. It has already been explained that the pressure spike in the hydraulic system, which is caused by the sudden stoppage of the hydraulic fluid masses, will probably not cause a raise in cavity pressure of sufficient magnitude and duration to cause significant deflections. Even in the presence of short lived pressure spikes, the magnitudes are so large that dies cannot be designed to remain unaffected by these loads. The dynamic response of the die includes a short-lived transient load that disappears very quickly. The clamping should be able to keep the dies from opening completely, but it cannot stop a dynamic response.

Pressure spikes should be dealt with in a different manner. From the point of view of the design of the die, one of the few alternatives would be to make modifications to the runner system to produce an increased back pressure as the cavity fills up, thus producing a favorable deceleration trace. Machine builders already provide their machines with plunger deceleration devices.

In this particular case, a hydrostatic pressure distribution based on a maximum cavity pressure (70 Mpa) was used. Given that there is no information as to how the cavity pressure changes as the part solidifies, it was assumed that the maximum pressure would hold for as long as the part remained in the cavity (16 seconds).

3.5.5 Other boundary conditions in the stress analysis model

Die casting dies are complex assemblies whose individual components are tightly fit. The most obvious boundary is the parting plane. Die builders make a great effort in making sure that there is an accurate match between the two die halves.

ABAQUS is capable of representing different types of surface interactions such as solid contact and friction. Three dimensional surfaces are defined as contact surfaces without the need for creating gap or interface elements. The presence of joints in the die assembly presents other modeling problems. For example, when compared to a homogeneous structure, a bolted joint produces a variation in the resistance to flow in the

path of heat. Also, the transmission of mechanical loads is clearly a function of the type of joint in question. In the case of the die, a large number of bolted joints are found. It is very difficult to economically model each joint explicitly in the type of simulation we are running.

Before the final model of the flat plate die was developed, the effects of the joints were tested. The results were reported by (Hegde 1995). This work showed that the deflections of the cavity surface were relatively insensitive to the presence (or absence) of bolts at the ejector shoe/support block interface. Clearly, stress/strain fields in the immediate vicinity of bolted are indeed affected. Due to their proximity to the region being analyzed (i.e. the cavity), bolts that hold the insert within the pocket of the ejector holder block were specifically modeled.

To constrain the model fully, the cover die was modeled as if it were clamped to a rigid foundation. This boundary condition is too stiff and more realistic modes will be addressed later in this report.

3.6 Simulation Results

Figure 3.5 illustrates the characteristics of the models developed for heat transfer and stress analysis of the flat plate die. Table 3.4 summarizes the cases that were simulated in this study. A full description of the results and raw data of this study are available in (Miller, Ahuett-Garza et al. 1995). In this table, *w* refers to a die with an insert and *wo* refers to a die without an insert.

Table 3.4: Process conditions for the different simulation trials

	Ejector Die thickness (t)		
	Dimensions in inches		
Load Conditions	4.875	5.875	6.875
70 Mpa (10,000 psi)	w/wo	w/wo	w/wo
70 Mpa (10,000 psi) + solidification heat	w/wo	w/wo	w/wo

Figure 3.5 Characteristics of models. (a) Stress analysis. (b) Heat transfer.

3.6.1 The first cycle

In the case of the flat plate die, deflections at the parting plane were chosen for analysis. More specifically, the parting plane separation at the edge of the cavity was observed for different cases. Separations at such locations may result in flashing

Figure 3.6 shows a comparison of the maximum parting plane separation as a function of ejector holder thickness. As can be observed, the maximum separation between die halves at the parting surface decreases as the thickness of the ejector holder shoe is increased. Also, the parting plane separations of the die with insert are larger than the

corresponding deflections of the monolithic die. This separation occurs between the overflows of the casting.

Simulation results show that, typically, the maximum separation starts at the instant the cavity pressure is applied. Figure 3.7 illustrates this phenomenon. After a short time, the heat load takes effect and causes the gap to collapse.

Figure 3.8 shows the variability of the cavity dimension. Before pressure is applied, the die has grown almost uniformly. This fact is the result of assuming that the casting cycle start after the die has been heated up uniformly to 100°C . The cavity space, measured in the direction of the thickness of the die half, has reduced slightly. As the pressure is applied, the size of the cavity is increases and, as the heat takes effect, the space is once again reduced. This recovery in the dimension can occur only if the part within the cavity can deflect under the pressure produced by this recovery.

Figure 3.9 illustrates the magnitude of the overall die growth after one cycle. This particular plot presents deflection data at a point on the back of the ejector holder. While the overall growth of the die is clearly dominated by the thermally induced strains, the magnitude of the instantaneous peak deflection depends upon the pressure being applied. In general, maximum deflections occur, near the middle of the die, away from the pillar support. Additional supports in this region would reduce the magnitude of the deflections.

3.6.2 Contributions by the different loads

From the perspective of the analysis of the behavior of the die, an important outcome of this exercise is the ability to visualize the relative importance of the different loads. Figure 3.10 shows the contributions of heat and pressure to the deflections of two points of the die. The deflections of the cavity can be used to explain the effects of each load. Before pressure is applied, the cavity is displaced from its original position by a significant amount due to the raise in temperature. The magnitude of this displacement is not presented in the figure because the size of the cavity remains almost unchanged. As pressure is applied within the cavity, the dimension measured across the thickness of the cavity grows larger. This is illustrated in the figure by the fact that the pressure induced deflection is larger than and opposite of the heat induced deflection. After a few casting cycles, the temperature of the die increases, resulting in a larger contribution of heat to the overall deflection.

A better picture of the contributions by each load is presented in Figure 3.11. This figure shows displacement fields, significantly exaggerated, at a plane that passes through the symmetry plane of the die.

Figure 3.11 (a) shows displacement fields after clamping force has been applied and prior to metal injection. Maximum deflections are of the order of 0.00005 m ($0.002''$) and occur at regions close to the pillar supports.

Figure 3.11 (b) presents displacement patterns due to the combination of clamping force and cavity pressure. No thermal effects have been included in the model in this case. Maximum deflections under these conditions are approximately 0.0003 m ($0.012''$) and occur approximately in the middle of the cavity. A gap is created at the parting plane in

Die with Insert

Monolithic Die
(cavity machined on die)

Case = Ejector Half Die
Ejector Half Die Dimensions

Height = $35 \frac{1}{2}$ in

Width = $23 \frac{3}{4}$ in

Thickness (t) = variable

Cavity Pressure = 10,000 psi

First casting cycle

Figure 3.6 Separation as a function of die thickness, cycle 1

Case = Ejector Die Half (Monolithic)

Ejector Die Dimensions

Height = $35 \frac{1}{2}$ in

Width = $23 \frac{3}{4}$ in

Thickness (t) = $4 \frac{7}{8}$ in

Cavity Pressure = 10,000 psi

First casting cycle

Approximate location of nodes

Magnitude of node displacement at opposite sides of parting plane, between overflows and adjacent to the cavity

Figure 3.7 Separation throughout the cycle

Case = Ejector Die Half (Monolithic)

Ejector Die Dimensions

Height = $35 \frac{1}{2}$ in

Width = $23 \frac{3}{4}$ in

Thickness (t) = $4 \frac{7}{8}$ in

Cavity Pressure = 10,000 psi

First casting cycle

Approximate location of nodes

Figure 3.8 Prior to the application of pressure, the cavity space measured across the thickness of the plate has shrunk slightly. As the pressure is applied the cavity space grows. Once the pressure is released, the space shrinks again.

Case = Ejector Die Half (Monolithic)

Ejector Die Dimensions

Height = $35 \frac{1}{2}$ in

Width = $23 \frac{3}{4}$ in

Thickness (t) = $4 \frac{7}{8}$ in

Cavity Pressure = 10,000 psi

First casting cycle

Approximate location of node
(Back of plate)

Figure 3.9 Overall growth of the die

Case = Ejector Die Half (Monolithic)

Ejector Die Dimensions

Height = $35 \frac{1}{2}$ in

Width = $23 \frac{3}{4}$ in

Thickness (t) = $4 \frac{7}{8}$ in

Cavity Pressure = 10,000 psi

Approximate locations

Contribution to maximum deflection by each type of load

Figure 3.10 Contribution of pressure and heat to deflection

the immediate region of the cavity. This gap is not seen in the proximity of the pillar supports.

Figure 3.11 (c) illustrates the effect of the heat load a few instants after heat is applied. In this case clamping force is applied, but no injection pressure has been introduced. The die tends to "bulge" in response to the stress caused by the temperature fields. If there were no mechanical stops behind the platen, the non uniform growth of the die would actually force the platen to move back until the stresses from the clamping force balance themselves with the stresses generated by the heat load. This phenomenon is shown by the fact that the maximum deflection (approximately 0.00035 m or 0.0015") is again found around the pillar support region, except that its direction occurs against the action of the clamping force. The gap at the parting plane is now in the periphery of the die, instead of in the middle of the cavity as in the cases in which only cavity pressure was applied.

Figure 3.11 (d) shows the combined effects of all loads: clamping force, injection pressure and heat from solidification. Maximum deflection occurs once again at the center of the die in a direction that tends to separate the die halves. The magnitude of this deflection is approximately 0.00051 m (0.020"). In this region, pressure (with 0.00026 m) and heat (with 0.00025 m) combine to produce the maximum deflection. The contribution of the clamping force is negligible in this region of the die. This figure also shows that a gap along the parting plane runs all the way from the cavity to the atmosphere. The results indicate that the effects of heat and pressure do not necessarily offset each other.

3.7 Multiple Cycles

The behavior of a die without insert during 4 cycles was simulated. In general, results show that the heat load tends to dominate deflections as the operation approaches quasi-steady state.

In this case, the heat load model was modified by reducing the amount of heat being released after each subsequent cycle. This is done to account for the fact that, as the temperature of the die increases, the heat that can be transferred from the part to the die is actually reduced for the same holding time. The models seemed to be fairly insensitive to the heat convection with the environment.

Figure 3.12 shows how the maximum parting plane separation keeps getting smaller as time progresses. In essence, the heat load begins to create a pre-stress at the parting plane as the die components try to grow due to the increased temperature. The location where the maximum deflection is found shifts from a region close to the overflows, to a point close to the middle of the cavity.

(a)

(b)

Figure 3.11 Effects of (a) clamping force, (b) clamping force and cavity pressure,

(c)

(d)

Figure 3.11 Effects of (c) clamping force and heat, and (d) all loads combined

3.8 *Characterizing the Long Term Behavior of a Die*

The picture that the flat plate results present is consistent with what is commonly observed in the field. In general, it is known that early in the casting operation dies tend to move a lot and in patterns that might be unpredictable. After several thermal cycles, deflections seem to converge to a regular pattern.

At the beginning of each casting cycle the stiffness of the die is defined by the design of the structure of the die, the bulk temperature, and the clamping force applied. The die temperature interacts with the die casting machine settings to determine the actual clamping force. At some point during the casting operation, initial die temperatures for each casting cycle will be approximately the same between consecutive cycles. Quasi steady state, or repeating periodic conditions, is reached at this point. A characteristic thermal distribution is reached within the structure of the die. This distribution combines with the fundamental design of the die and the properties of the die material to define the susceptibility to deflect.

This behavior is illustrated by the results shown in Figure 3.12. This figure tracks the z displacement of the two nodes located at a similar position on either side of the parting plane (i.e. one node is on the cover side while the other is on the ejector side). At the beginning of each cycle, both nodes are coincident (dies are closed). As pressure in the cavity is applied the nodes separate, indicating that the die halves move apart. Once heat from solidification takes effect the dies tend to grow. As a result both die halves try to come together again. This sequence repeats itself in each cycle.

Figure 3.12 clearly shows how the starting position for each cycle keeps shifting. Similarly, the maximum separation between consecutive cycles keeps decreasing. Field observations confirm this behavior. Figure 3.13 shows a schematic diagram of the placement of displacement probes on the structure of a die for a transmission casing⁷. Figure 3.14 shows that the maximum parting plane separation at an outside corner of the die changes slightly over a period of 40 cycles. These data were collected during a period that was considered to represent quasi steady state conditions. Process conditions remained fairly constant during this period of the casting operation. In Figure 3.15, the separation at an outside corner of the flat plate die shows the same characteristic behavior.

Figure 3.15 also contains results from a model using spring elements to roughly account for the flexibility of the machine. The magnitude of the deflections increase, but the pattern remains essentially the same. This suggests that the transient behavior is inherent in the system and is due primarily to geometric effects, i.e., the changes in the shape of the die due to non-uniform temperature fields that are changing from cycle to cycle in the transient period.

⁷ Data courtesy of Matrix Technologies, Muncie In.

Figure 3.12 Parting Plane separation at the midpoint of the plate

Figure 3.13 Schematic of location of displacement sensors on a transmission case die. Reproduced with permission of Matrix Technologies, Inc.

(a)

(b)

Figure 3.14 (a) Parting plane separation as reported by Matrix (b) Process conditions (scaled for display)

Figure 3.15 Schematic comparison of die support. (a) Original model, (b) Modified model accounting for elasticity of supports, (c) Comparison of simulation results, outside corner of die.

3.9 Relevance of Simulation Control Parameters

The choice of simulation control parameters has a very important effect upon the quality of the results and the cost of the simulation exercise. The role that the simulation control parameters play is illustrated by the difference in results between a single cycle and a multiple cycle simulation.

When compared to a single cycle simulation, simulating multiple cycles requires the modification of certain simulation control parameters. For example, because more time is being simulated, the user is forced to decide the frequency with which data is to be collected and filed during the numerical computations. This has an important effect upon the values of the output data, because peak values may be missed by not storing data frequently enough. A conflicting constraint is presented by the fact that more frequent storage increases the size of the results files.

3.10 Sensitivity and Accuracy of Results

Simulation accuracy is affected by the load magnitudes, the choice of boundary conditions and the quality of the mesh. It is important to emphasize that given that the final goal of a simulation is to reduce operational problems and provide for dimensional control of the part, results need to be accurate to a scale that is significant for these purposes. In general, allowances in the 0.0001 m (0.005") range are considered tight for die casting.

Based on the approach described in this chapter, accuracy of simulation results will fall within this range. For example, experience with the flat plate die indicates that deflection results are fairly insensitive to variation in the cavity temperature up to about 70 °C. The effect of variations of up to 30 °C in the bulk of the die has also negligible effects.

Finally, a clear source of inaccuracy in the model for the flat plate die is the manner in which the support from the die casting machine is modeled. As has been mentioned, by fully clamping one side (cover die) and applying a uniform pressure on the other die half (ejector), the elasticity of the support that the machine provides is totally ignored. The flexibility of the die casting machine is considered in more detail in chapter 5.

3.11 Conclusions

A methodology for the simulation of die casting die deflections has been developed. The issues in this process, and recommendations for addressing them, were also explained. The results seem to be in the range of what is commonly observed in the field.

In the next chapter these procedures and recommendations will be applied to several case studies and the modeling approach will be refined further to better account for the complex die casting process.

4. MODEL DEVELOPMENT AND VALIDATION

4.1 *Horn Pin Lock*

4.1.1 Background

Auto components presents some of the most dramatic examples of die deflections. Currently, most of the large components of the drive train are die cast. Because of their geometric characteristics, the manufacture of these parts requires the introduction of moving die members. In general, slide deflection magnitudes are not negligible. For example, data collected in the field has shown that slides may blowback as much as 0.035" during operation. Compensating for this deflection has the potential to significantly reduce production problems and costs.

4.1.2 Objective

The objective of this study was to develop models to predict slide deflections. Due to the availability of field data from the die manufacturer (Exco Engineering), a horn pin type lock was selected for this study. Figure 4.1 shows the rough schematic of the geometry and dimensions of the horn-pin lock against which models were designed and tested. Figure 4.2 presents a blowback trace for this slide as collected in the field.

4.1.3 Simulation Procedure

As in previous work, IDEAS (SDRC) was used for preparation of the finite element models and ABAQUS (Hibbit, Karlsson & Sorensen, Inc.) was used for carrying out numerical computations and displaying simulation results. Figure 4.3 presents a schematic of the model with its boundary conditions. Assuming symmetry about the mid-plane between horn pins reduced the size of the problem. Consequently, only half of the geometry was actually built. Strictly speaking, the geometry is not symmetric. However, given that mechanical loads upon the slide tip were reduced to a hydrostatic pressure, the only source of unbalance was the projected area of the slide tip that was calculated to be 0.0642 m^2 . As the field data proved, slide tilt is relatively small. In other words, the slide moves almost straight back and asymmetry is not significant.

In order to account for the effects of the support from the cover shoe, a small support block was modeled at the bottom of the pin. Contact surfaces, a numerical feature with which model interfaces, were used to represent interfaces between pin and slide. Displacement of the back of the slide matched field data when a friction factor of 0.3 was used. Figure 4.3 (b) shows typical simulation results for these conditions and a cavity pressure of 95 MPa. For detailed modeling procedure see reference (Ahuet-Garza 1996).

4.1.4 Results and Discussion

Figure 4.3 (b) shows blowback results for Exco's original lock design. Simulation results report a blowback of around 0.036" (9.195 E-04 m), which falls within the range displayed by field data confirming that the modeling approach is capable of predicting die deflections.

Figure 4.1 Overall dimensions of Exco's slide carrier and pin

Trace of average slide blowback
Channels 1 & 3, Timing for Intensification = 0.5 sec.

Figure 4.2 Slide blowback. Pan Slide, Exco's design

Figure 4.3 Description of model for blowback computation

4.2 Matrix - Delaware: Combination Lock

Another lock design was provided by Delaware Machinery. Figure 4.4 shows a schematic of the geometry and the dimensions of this lock design. In essence, this lock is a combination of horn pin and inboard lock (notice the extra support provided by the cover die). In this case a block was added on the top of the slide in addition to the block at the bottom of the pin. A feature of this particular design is that locks are interchangeable. For this reason, a clearance of 0.004" is provided to facilitate assembly.

Figure 4.5 shows the boundary conditions and the results for the above design for a cavity pressure of 71.8 MPa. Simulated blowback in this case is 0.028". Because the model does not handle rigid body motion, assembly clearances must be added to computer total blowback. Still, simulation results fall short of actual field data collected by Matrix Technologies. Figure 4.6 shows the measured value to be approximately 0.036".

When this lock was simulated without the extra support provided by the cover die, the results match the field data. This suggests that the combination lock may not perform as originally intended.

Figure 4.4 Overall dimensions of pan slide carrier (inches)

Figure 4.5 Schematic of the Delaware slide pin with boundary conditions

Figure 4.6 Slide blowback. Pan slide, Delaware Machinery design

4.3 Models for Proud Insets

The objective in this section is to identify parameters that can be used to evaluate die performance. Models that address the contributions that proud inserts (inserts held a few thousands of an inch above the surface of the die) make to the performance of die are used for this purpose.

4.3.1 Models and boundary conditions,

Inserts maybe placed in the die shoe in one of two ways:

1. flush with the die surface at the parting plane.
2. protruding a few thousands from the die surface at the parting plane.

It is believed that when the inserts are allowed to protrude from the pocket, a higher preload is generated as compared to the case with flush inserts.

Figure 4.7 shows the flat plate die discussed in chapter 3 arranged to account for proud inserts.

Simulations with two types of models were carried out.

Model 1 - Rigid support at the back of the cover die.

Model 2 - Spring elements approximate the effect of the cover and ejector platen.

In the first case, by fully clamping the cover die and applying a uniform pressure on the ejector, the elasticity of the support that the machine provides is totally ignored. The second model accounts for this factor by using spring elements to model the platen support behind the dies. These two types of support are shown in Figure 4.8. The equivalent stiffness of the springs is computed by calculating the stiffness of different components and then calculating an overall stiffness for the series/parallel arrangement.

Tables 4.1 and 4.2 contain the process conditions and characteristics of the casting cycle used in the simulations.

A heat transfer analysis was carried out to compute the thermal fields in the die. The thermal loads are computed by first simulating the die geometry in MAGMA and using the outputs of this simulation to estimate the heat flux. Based on the results of simulation with MAGMA, five different heat flux regions were identified within the casting. Appendix B discusses in detail the procedure involved in calculating the heat flux values for a similar casting.

The heat transfer model for the flat plate die accounted for convection and conductance between the different interfaces. It was assumed that convective heat transfer takes place at the cooling channels and the exterior of the die during the die holding time, lubricant spraying and die open time. When the die is closed, the parting plane surfaces are in contact and their interface is represented by a finite conductance in the heat transfer model. When the die is open, these surfaces are exposed to the atmosphere, and therefore a convection heat transfer boundary condition is added.

Clamping forces were modeled as a uniform pressure on the back of the die. Cavity pressure was assumed to be hydrostatic pressure applied over the element faces in the cavity. In this particular case 70 MPa was applied.

Table 4.1 Process Conditions

Casting Material	Al 380
Melt Temperature	600 degrees C
Injection Pressure	10,000PSI
Total Cycle Time	36 sec
Clamping Force	815 tons
Ambient Temperature	20 degrees C
Heat transfer coefficient Steel – air	20 J/degree K
Heat transfer coefficient Steel – steel	2000 J/degree K
Water temperature	40 degrees C

Table 4.2 Casting Cycle

Time (sec)	Step
0-1	Dies Clamped
1-16	Material is injected and pressure applied to cavity
16-17	Die opened and part removed
17-27	Die idle
27-30	Lubricant spray
30-35	Die idle
35-36	Die clamped

Figure 4.7 Geometry of the die and part

Figure 4.8 Rigid and spring models

4.3.2 Model with Rigid support at the back of the cover die

The boundary conditions for this model are:

- Cover die is fully clamped at its back.
- Clamping pressure corresponding to a clamping force of 800 tons is applied at the back of the ejector support.

The boundary conditions are shown in Figure 4.8a.

The two cases that were analyzed are:

- Inserts flush with the die block (flat parting plane)
- Inserts stick out from the die block by 0.004" (proud inserts)

Eight casting cycles were simulated. Contact pressures were noted when the dies were clamped. The contact pressures are an indication of the preload at the parting plane. Parting plane separations were recorded at when maximum cavity pressure was applied in each cycle. The nodal locations where contact pressure and separation were recorded are situated along the edge of the cavity as shown in Figure 4.9.

Figure 4.10 shows the comparison of simulation results for the average contact pressure in the vicinity of the cavity and the change in the contact area at the parting plane between two cases. In the first cycle the contact pressures are higher for the case with proud inserts, because clamping force is concentrated on a smaller area. The contact pressures for both cases increase with each cycle. This is due to the fact that while the clamping force remains constant, the contact area at the parting plane decreases due to thermally induced distortion of the die.

Figure 4.11 shows the simulation results for average and maximum parting plane separations. Initially both the average and maximum separation for the case with proud inserts is smaller than for the case with flush inserts. As seen in the figure the magnitude and patterns of separation for the two dies become more and more similar as the number of cycles increases.

Figure 4.9 Locations where separation and contact pressures were recorded

Figure 4.10 Simulation results for average pressure and variation of contact area, rigid model

Figure 4.11 Simulation results for average and maximum separation, rigid model

4.3.3 Model with spring elements to include the effect of platens

Simulations using the same geometry were performed by adding spring elements to account for the flexibility of the support provided by the cover platen, ejector platen and toggle linkage. Figure 4.8 contains the schematic diagram of the model with spring elements. A uniform spring distribution is placed behind the cover die and ejector die. Displacements were applied at the junction of the ejector platen and toggle linkage to produce a force equivalent to 800 tons. The two cases analyzed are the same as before:

1. Inserts flush with the die (flat parting plane)
2. Inserts sticking out by 0.004" (proud inserts)

The simulation results show, Figure 12, that the contact pressures increases for both cases with each cycle due to a decrease in the contact area at the parting plane. As shown in Figure 4.12 the contact area for both cases is essentially the same after cycle 5. Figure 4.13 shows the comparison of average and maximum separation for the two cases. There is almost no difference between the cases.

4.3.4 Discussion of Results

It is observed that the separations for the spring model (Cases 3 & 4) are higher than those for the rigid model (cases 1 & 2). This is one more indication that machine flexibility is important.

As would be expected, the contact pressures present the opposite behavior. Contact pressures are higher for the rigid model. A more rigid system will hold the die together better.

The results from both models indicate that the effect of proud inserts is more significant early in the operation, but as the dies warm up their effect decreases.

The use of proud inserts may address some manufacturing/assembly concerns, but it probably does not have much effect on flashing.

For details on this modeling procedure see reference (Dedhia 1997).

4.3.5 Preliminary study of the sensitivity of die design variables on die deflection

A preliminary study was conducted to identify important design parameters. The study consisted of analyzing the effect of variation in the thickness of the die and locations of the ejector supports on die performance. The model that was used for simulations is the same die used for the analysis of proud inserts. The spring elements were used to include the effect of cover and ejector platen, toggle linkage. Displacement boundary conditions were used at the junction of the ejector platen and toggle linkage. These displacements produce a force equivalent to 800 tons.

The measures that were recorded are:

- Maximum separation at the parting plane
- Average separation at the parting plane
- Maximum contact pressure at the parting plane
- Average contact pressure at the parting plane.

Figure 4.12 Simulation results for average pressure and variation of contact area, spring model

Figure 4.13 Simulation results for average and maximum separation, spring model

The locations where the above responses were collected are shown in Figure 4.9 and correspond to those used for the analysis of proud inserts. The thickness of the cover and ejector die were varied and their effects analyzed. Three cases that were analyzed are shown in Figure 4.14. Eight casting cycles were simulated, each cycle lasting 36 seconds. Contact pressures were recorded at die lock up. Separations were recorded when intensification pressure is applied in each cycle.

Figure 4.14 shows the comparison of contact pressures and separation in the vicinity of the cavity for the three cases that were analyzed to study the effect of die thickness. The maximum contact pressure and average contact pressure both increase with each cycle. The average contact pressure is higher for the case where the dies are thicker than the nominal case. Correspondingly, the maximum separation and average separation is lower for thicker dies when compared to the nominal case.

4.4 Summary

The case studies presented above serve several purposes. First and foremost, the lock analyses show that the modeling approach is capable of describing actual performance. The simulation results and field data match quite well.

Second, the results strongly suggest that the flexibility/rigidity of the machine structure must be accounted for when modeling die deflection. It has been shown that parting plane separation is sensitive to the assumptions about the rigidity of the machine. This result is not surprising, but it does mean that more complex and more complete models are required.

Finally, the results suggest that parting plane separation, both average and maximum, is somewhat more sensitive to changes in design variables (e.g., thickness) than are average or maximum contact pressure. For this reason, parting plane separation measures will be used as the measure of effectiveness in the design guideline case studies presented in chapter 6.

Locations near the cavity edge

	CDT Cover die thickness	EDT Ejector die thickness	EST Ejector support thickness
Thin	5.875"	4.875"	6.0"
Medium	6.875"	5.875"	6.0"
Thick	7.875"	6.875"	6.0"

Figure 4.14 Comparison of 3 die thickness

5. ACCOUNTING FOR THE DIE CASTING MACHING

A study of the effects of the die casting machine on die deflections is presented in this section. Ultimately, the intent is to account for the machine through the use of either spring elements or an alternate model, thus avoiding the need to explicitly include the geometry of the entire machine in the models of die deflections. This requires a thorough understanding of the support that the machine provides for the die. To develop this understanding, FE models that simulate the clamping action of the die casting machine were developed. A case study and experimental verification of the results are presented in this chapter. This chapter is based on Aswin Choudhury's master's thesis. More details about the study can be found in (Choudhury 1997)

5.1 *Full Machine Model and Verification*

5.1.1 Experimental data

Platen deflection data was obtained for a 700-ton cold chamber die casting machine with the cooperation of Prince Machine Co. The experimental setup consisted of a beam attached to a mounting fixture that sat on the floor. The machine was not anchored to the floor. The structure was placed near the cover platen. A dial indicator was mounted on the beam. Another dial indicator was mounted on the foundation near the base of the cover platen to verify its displacement. Lock up of the machine was done very slowly. Platen deflections were recorded at eight different locations on one half of the platen as shown in Figure 5.1. Four readings were taken for each location. The machine had to be opened and closed to record each individual measurement.

Two points on the other half of the platen were included in the measurements to confirm the symmetry of the deflection pattern of the platen (Figure 5.1). The results suggested that platen deflection is nearly symmetric.

Details of the machine construction that were relevant for modeling that were confirmed during a trip to Prince were:

- The base frame on which the machine is placed is not bolted to ground
- Cover platen is bolted at the bottom to the base. It moves by .0005" to .001" towards the shot sleeve during clamp
- Ejector platen is squeezed between the die and the toggle linkages
- The C-frame is bolted to the base
- Material used for platens is 1018 steel
- Material used for the tie bars is 4340

A separate set of platen deflection data was collected with a Lieca Laser Tracking System from a similar 700-ton machine in June 1997.

Three sets of data are shown in Table 5.1 (Machine 1), Table 5.2 (Machine 2) and Table 5.3 (Machine 3). The locations where data were collected are shown in Figure 5.1. Deflection patterns for the three cases are shown in Figure 5.2.

There is significant variability in the deflection patterns and values for the three machines. The data for machine 1 in particular differ from the other two. The difference in deflection values at location 5 is around 0.004". This difference may be due to the assembly of the machine, difference in stiffness of the components of the machine, inaccuracies in the calibration of the instruments used for the measurement, unbalanced load distribution in the structure or a combination of all these factors.

Figure 5.1 Locations on front of cover platen where data was collected.

Location	Deflection (inches)		
	Meas. 1	Meas. 2	Meas. 3
1	0.0090	0.0095	0.0095
2	0.0090	0.0090	0.0090
3	0.0075	0.0080	0.0075
4	0.0065	0.0075	0.0065
5	0.0030	0.0030	0.0030
6	0.0170	0.060	0.0160
7	0.0150	0.0150	0.0150
8	0.0125	0.0120	0.0120

Table 5.1 Measured deflections, Machine 1

Location	Deflection (inches)			
	Meas. 1	Meas. 2	Meas. 3	Meas. 4
1	0.0100	0.0100	0.0095	0.01000
2	0.0105	0.0105	0.0105	0.0105
3	0.0085	0.0085	0.0085	0.0085
4	0.0090	0.0090	0.0090	0.0090
5	0.0075	0.0075	0.0075	0.0075
6	0.0130	0.0130	0.0130	0.0130
7	0.0130	0.0130	0.0130	0.0130
8	0.0115	0.0115	0.0120	0.0120
9	0.0110	0.0110	0.0110	0.0110
10	0.0110	0.0110	0.0105	0.0110

Table 5.2 Measured deflections, Machine 2

Location	Deflection (inches)
1	0.0114
2	0.0115
3	0.0088
4	0.0106
5	0.0078
6	0.0144
7	0.0137
8	0.0129
9	0.0172
10	0.0186

Table 5.3 Measured deflections, Machine 3

Figure 5.2 Behavior of deflection pattern shown by three 700-ton machines

Note

Machine 1: Data received on 1/7/97.

Machine 2: Data collected during our visit to Prince.

Machine 3: Data received on 6/26/97.

5.1.2 Model of die casting machine

Figures 5.3 and 5.4 show the dimensional characteristics of the dummy die and the components of the machine that were simulated. The dummy die is a structure that consists of two plates welded with four pillars between them. As described above, the die casting machine modeled is a 700-ton machine.

Much attention was paid to the modeling of the cover platen to include the details such as the T-slots, shot sleeve and the holes near the C-frame as shown in Figure 5.5. The ejector and rear platen were modeled as simple plates with holes that guide the tie bars. A group of I-beams were used to model the C-frame. Due to the symmetry of the problem, only one-half of the geometry was used for simulation.

Figure 5.3 Schematic and dimensions of the dummy die

5.4 Schematic of the components of the die casting machine accounted for in the model.

Figure 5.5 a) Front of the cover platen showing the details of the model
b) C-frame modeled as a group of I-beams.

5.1.3 Simulation conditions

The boundary conditions for the dummy die and the machine are shown in Figure 5.6. In the experimental setup, grease was applied to the front of the cover platen to avoid damage to the platen and the dummy die was not clamped to this platen. To model this, a contact surface with a zero friction coefficient was applied at this interface. The dummy die was bolted to the ejector platen at four locations (see Figure 5.7). The nodes in the ejector platen and dummy at corresponding locations were tied together to represent the bolts.

In the experimental setup, the load on each tie bar was equal to 175 tons. For the application of displacement boundary conditions, the theoretical elongation on each of the tie bars was calculated using the procedure described by Herman (Herman 1988). The theoretical displacement obtained was divided and applied to the back of the support blocks on the ejector and rear platens. The displacements were adjusted to achieve the required tonnage.

The total displacement applied on the back of the ejector and the rear support block was divided in the ratio of 40:60 respectively to match the simulation results with field data.

The loads in all tie bars were matched. The results indicate that a total displacement of .054 in. and .052 in. applied at the top and bottom toggle support block respectively produces a total tie bar force of 700 tons as shown in Figure 5.8. No cavity pressure or heat loads were included in this model since the experimental data were taken under die lock up only.

Figure 5.6 Boundary conditions used in the simulation.

Figure 5.7 Location of clamps on the ejector side.

Figure 5.8 Simulations results showing tie bar loads at lock up

5.1.4 Results

Figure 5.9 (b) summarizes the deflection pattern obtained from the simulation compared with two sets of field data. The location of data points and the two cross sections where data were collected are shown in the Figure 5.9 (a).

Simulation results for locations 1, 2, 3, 4, 6, 7 and 8 were within 10% and location 5 was within 15% of experimental data for machines 2 and 3. Results at locations 1, 2, 3, and 8 were close to experimental data for machine 1. However results differ considerably from field data at locations 4, 5, 6 and 7 for machine 1.

In general, deflection patterns match the field data well. The predicted deflection values were within 10% - 15% of the field data. This discrepancy is small considering the large number of variable factors that needed to be included in building the model. The full machine model can be used as to produce benchmark data for the development of an alternate model to account for the die casting machine in die deflection simulations.

The deflection patterns and values are very sensitive to the boundary conditions applied at the bottom of the cover platen and C-frame. The best results in terms of matching the measured deflection patterns were obtained using roller support as the boundary condition at the bottom of cover platen.

Figure 5.9 a) Platen deflection locations. b) Comparison of field data with simulation results at cross section A and B.

5.2 Comparison of Models

It was shown in the previous section that a good model of the full die casting machine can be constructed, but this model is quite large due to the size of the machine. There is a need to simplify the model to produce a model that is still accurate but more computationally tractable. The first step is to generate results for die separation using a full machine model and compare these results with the analyses described earlier that assumes a rigid machine.

Figure 5.10 shows the schematic of the die and the machine used for simulations. The machine considered is an 800-ton die casting machine. The die modeled was an open/close die with dimensions as shown in Figure 5.11 (a). This die covers 65% of platen area between the center of the tie bars. The casting used for this set of simulations is a box type with dimensions as shown in Figure 5.11 (b). The following cases were simulated for a single cycle considering machine clamp and intensification pressure. Heat was not included.

Case 1 - The die was modeled along with the full machine. The boundary conditions used for this case were same as defined in the previous section except that the cover die was clamped to the cover platen at four different locations (Figure 5.12).

Case 2 - This is the model described in chapter 3. Figure 5.13 (a) shows the boundary conditions used for this case. The cover platen provides rigid support. All the nodes on this surface were fixed. The total clamping tonnage was applied as an equivalent pressure on the ejector support.

Case 3 - Spring-elements were placed behind the cover and ejector dies as shown in Figure 5.13 (b). On the cover side, spring elements were used to account for the flexibility of the platen. On the ejector side, two different sets of spring elements were used. One represents the ejector platen and the other one represents the toggle/tie bar arrangement. In addition, a clamping force was applied at the toggle / platen junction. The equivalent stiffness of the springs was estimated by calculating the stiffness of the different components and then calculating an overall stiffness for the series / parallel arrangement.

Maximum separations at the parting plane are shown in Figure 5.14. Although the pattern of the parting plane separation for all the cases is reasonably similar, the magnitudes are quite different at most locations. The parting plane separation for Case 1 i.e., the case including the whole machine is 150% than Case 2 and 100% higher than Case 3. In fact, the stiffness of the springs in Case 3 had to be reduced to 10% of its original value to produce results equal to that of Case 1 (see Figure 5.15). This argues that it is the localized support of the platen due to the tie bar locations that reduce support for the die. Case 2 essentially describes a machine that has a uniformly distribute elastic foundation. Case 1 has localized support from the tie bars.

Figure 5.10 Schematic of the components of the die casting machine and machine specifications for an 800 ton machine.

Figure 5.11 a) Schematic showing the geometry of the die used in the simulation. b) Dimensions of the part selected for an 800-ton machine.

Figure 5.12 Diagram showing the boundary conditions used in the simulation when the die modeled with whole the machine.

Case 2: Rigid Machine

(a)

Case 3: Die with spring elements

$K_1 = \text{Cover platen stiffness}$
 $K_2 = \text{Ejector platen stiffness}$
 $K_3 = \text{Toggle / Tie Bar stiffness}$

(b)

Figure 5.13 a) Boundary condition for Case 2 rigid machine model. b) Boundary condition for Case 3 (model that accounts for elasticity of support by spring elements).

Figure 5.14 Comparison of simulation results for three different cases.

Figure 5.15 Comparison of results for Case 3 using different stiffness values.

Figure 5.15 provides a powerful illustration of the fact that the flexibility of the machine influences die deflections. Note also that the patterns of the curves are not quite the same as the full machine model case. Deflections tend to be higher at the top for case 2 in comparison to the full machine model. This is additional evidence that the pattern of support cannot be uniform and further justification for the spring/platen model.

5.3 The Spring/Platen Model

Modeling the whole die casting machine is a very time consuming process. The total modeling time would be approximately 2 days and the time required to run the simulation with proper resolution and multiple cycles would be around 18 hr. of CPU time without

heat effects and considerably longer if heat is considered. As described above, the difficulty with modeling the die with spring elements is determining the distribution and stiffness of the springs if the case 2 model is used.

In order to avoid these problems, a spring/platen model was developed to account for the machine as shown in Figure 5.16. Instead of modeling the whole machine, only the cover and ejector platen is explicitly modeled. Figure 5.16(b) shows the boundary conditions used. A constant pressure is applied at the back of the toggle support block on the ejector side to represent an equivalent clamping force. Spring elements are used at the back of the cover platen at the locations where the tie bars are attached to the platen to represent the tie bar stiffness. Contact surfaces were used at all the interfaces. Because the effect of heat is not included in this analysis, the die growth due to thermal loads is not included in the following results.

Figure 5.17 shows the comparison of maximum separation obtained for the spring/platen case and when the die was modeled with the full machine. The percentage difference in parting plane separation between the two cases for all the locations is plotted. Deflection results show that the pattern remains essentially the same. Also the deflection values differ by only 6% to 7% at any location.

5.3.1 Effect of C-frame force

A separate model was designed to find the force generated by the C-frame on the cover platen during the intensification. This force indirectly effects the parting plane separation. Figure 5.18(a) shows the FE model of the C-frame and the cover platen. A roller support was applied at the bottom of the cover platen and an equivalent cylindrical force was applied at the location shown in the figure. As shown in Figure 5.19, simulation results indicate that the resultant force due to C-frame on the cover platen is around 3.7×10^4 N (top) and 6.5×10^4 N (bottom) at the instant of intensification.

In order to check whether this force has any effect on the parting plane separation, a simulation was run by applying this force in the full machine model at the locations shown in Figure 5.19 (b). The results show that the force generated by the C-frame during intensification has minimal effect on the cover platen deflection (Figure 5.20).

For the purpose of modeling the support provided by the machine to the dies, the effect of C-frame can be neglected. If one wants to include this effect in the model the above procedure can be followed.

Figure 5.16 a) Schematic of the components of the spring/platen model b) Boundary conditions

Figure 5.17 Simulation results for full machine and spring/platen models

Figure 5.18 a) Geometry and boundary conditions for C-frame and cover platen.
b) Locations where force due to C-frame is applied.

Figure 5.19 Simulations results showing stress in the C-frame.

Figure 5.20 Comparison of results with and without the C-frame

5.3.2 T-slots

To check the effect of T-slots, a set of simulations of machine lock up was run. In one case the cover platen was modeled with the T-slots and in the second the T-slots were included. The platen deflections obtained for both the cases are shown in Figure 5.21. The deflections with T-slots are about 4% more when they are ignored. This suggests that the T-slots can be ignored in most instances.

Figure 5.21 Graph showing comparison of results for the cases when machine modeled with cover platen with and without t-slots.

5.4 Consideration of Different Part geometries

To verify the spring/platen model two additional parts were modeled. The two cases were:

Part 1 - the flat plate die used in chapter 3 was modeled in an 800 ton Prince die casting machine.

Part 2 - A two-cavity box type part was modeled in a 1000 ton Prince machine. The basic die geometry and the nodal locations where data were collected are shown in Figure 5.21 (a) and Figure 5.21 (b) respectively.

Figures 5.25 and 5.26 show the locations of nodes where data was recorded and comparison of results obtained for the above two cases.

The results show that the difference in predicted maximum separation at the parting plane is around 6% to 8% for these two models. The pattern is also very similar. The spring/platen model seems to be a good compromise between accuracy and computational complexity and can be used for die deflection analysis.

One extension that must await future work is approximating the compliance of the toggle mechanisms with springs. Without this addition, the spring/platen model cannot accommodate the increase in load to the die/machine structure that takes place with heat build up over cycles. The model as presented corresponds to the situation in which the tie bars are adjusted from shot to shot in order to keep the loads constant.

All of the tests to date have been with open-close dies. These dies have relatively balanced loading. Another extension for future work is expanding the model to account for lateral and vertical support provided by the tie bars and foundation when the loading is not as uniform. These loads may occur with badly designed dies and, possibly, with slides.

Figure 5.22 a) Schematic of the two-cavity die. b) Geometry of the casting.

Figure 5.23 a) Nodal locations where maximum separation was recorded
b) Comparison of simulation results for flat plate die.

Figure 5.24 a) Nodal locations where maximum separation was recorded
b) Comparison of simulation results for two cavity die.

6. STEPS TOWARD DESIGN GUIDELINES

In this chapter, several examples/case studies illustrate the basic use of the modeling approach developed and provide a start toward the development of design guidelines for die casting dies that take die deflections into account.

6.1 *Parametric Study of Horn Pin Locks*

6.1.1 Procedure

Once a working model was developed, an exercise was designed to study the effects of variations in the dimensions of selected parameters upon the magnitude of slide blowback and force transmitted into the cover die. Figure 6.1 describes the horn pin geometry and dimensions that were varied in this parametric study (Ahuett-Garza 1996). An FEM model was built for each case described in the table shown in Figure 6.1. Instead of reporting only maximum blowback for a given condition, stiffness of the pin and the force transmitted were considered as the parameters with which a particular design was to be evaluated. A larger stiffness results in a smaller blowback, and consequently is a desired characteristic of the design. In all cases the stiffness proved to be constant for the range of loads selected (20 - 40 MPa, or about half of maximum intensification pressure). On the other hand, a larger force transmitted into the cover die results in the need for a higher clamping force, and is therefore an unwanted result of a given modification to the original design.

6.1.2 Results and discussion

The chart in Figure 6.2 summarizes the results of the parametric study of Exco's lock design. In the graph, the value of stiffness and the force transmitted to the foundation of the original case were used as reference, and therefore a value of 1 was assigned to each one of them. For example, a stiffness of 1.43 represents a 43% increase over the original case. A third parameter, stiffness/force is also included in the chart. As shown by the chart, force transmitted to the foundation is more sensitive to changes in the design of the lock than stiffness. Only case 5 seemed to present a larger stiffness, at the expense of higher force into the cover die. However, the difference is small enough to be considered within the range of error for the modeling scheme. Case 4 is interesting because it falls outside this range, and it seems to reduce forces into the over die more dramatically than the reduction in stiffness it produces.

The most interesting aspect of this study is the fact that this lock design has evolved over a number of years. The part is high volume and a number of dies have been manufactured. Over time the design parameters have been modified by trial and error to the nominal values considered. The computational experiment basically confirms that the empirical design process has produced a design that is optimal with respect to stiffness and minimum force transferred to the cover die. This is encouraging and suggests that the computational approach can achieve similar results in new designs without the trial and error.

Parametric Study of Exco's Horn Pin Design

Parameters and Variations

Where:

$t/w (O) = 1.964$
 $t/w (H) = 1.3 \times t/w (O)$
 $t/w (L) = 0.7 \times t/w (O)$

$\alpha (O) = 3.5 \text{ deg}$
 $\alpha (H) = 1.3 \times \alpha (O)$
 $\alpha (L) = 0.7 \times \alpha (O)$

$\beta (O) = 10 \text{ deg}$
 $\beta (H) = 1.3 \times \beta (O)$
 $\beta (L) = 0.7 \times \beta (O)$

Constraint:

Area of top of pin
 $t \times w (O) = t \times w (H)$
 $= t \times w (L)$

Case	t/w	α	β
Original	O	O	O
1	H	H	H
2	H	H	L
3	H	L	H
4	H	L	L
5	L	H	H
6	L	H	L
7	L	L	H
8	L	L	L
OHH	O	H	H
OHL	O	H	L

Figure 6.1 Description of cases for parametric study of horn pin designs

Figure 6.2 Comparison of characteristic behaviors of slide/lock arrangements

6.2 Parametric Study of an Open-Close Die

More details on this work are reported in Sanjay Dedhia's masters thesis, (Dedhia 1997).

6.2.1 Approach for the sensitivity analysis

The NADCA Computer Modeling Task Group recommended the design variables and the part geometry used for this analysis. The characteristics of the analysis are described below.

- The casting was selected to fit in an 800 ton machine.
- The size of the die was varied from 50% to 80% of the platen area within the center of the tie bars as shown in Figure 6.3.

- The variables defining the structural characteristics of the die were varied to test the sensitivity to each variable.
- A response surface for the features of the die that are most sensitive to die deflection was created

Figure 6.3 Platen area within center of tie bars

The part selected for the analysis is a box shaped part as shown in Figure 6.4. The dimensions of the part were selected such that the projected area of the part would generate a separating force of 616 tons for a cavity pressure of 10,000 psi. The design variables are shown in Figure 6.5. They are:

- Die size (DS)
- Cover die thickness (t1)
- Ejector die thickness (t2)
- Insert thickness (t3)
- Circular post diameter (d)
- Proud inserts

An experimental array was designed in order to analyze the effect of design variables. This array was used in identifying important die design variables and generating response surface for the above variables.

Figure 6.4 Box casting

6.2.2 Deflection analysis for the design cases

Heat loads computed as described in Appendix B were used in the stress/deflection analysis of the dies. The spring/platen model (Figure 5.16) discussed in the previous chapter was used for the deflection analysis. Clamping pressure corresponding to a force of 800 tons was applied at the toggle support blocks at the back of the ejector platen.

The parting plane separation at the insert face was recorded. Figure 6.6 shows the insert faces at the parting plane where the maximum separations were recorded. Ten die casting cycles were simulated. Maximum parting plane separation was found along the outer edge of the insert as shown. The maximum separation occurred at the same location in all the cases. The location of the maximum parting plane separation is not surprising since the heat load causes thermal growth near the cavity edge causing the die to bulge outward resulting in maximum separation in that region. Figure 6.7 shows the cases analyzed and Figure 6.8 shows the maximum separation recorded for all the cases.

Figure 6.5 Variables used in the study

Figure 6.6 Insert faces where separation was recorded

6.2.3 Contribution of structural variables to die deflection

Dies are assembled from a number of components and varying the size of the structural components affects the stiffness of the die. An experimental array was designed as shown in the Figure 6.7 for studying the effect of the variables defined above and their interactions. This approach was adopted so that the desired information could be obtained with minimum number of simulations and for studying interaction effects between two or more factors.

As discussed earlier maximum separation was used as the response variable and regression equation was developed based on the results obtained from the simulations. This regression equation could be used to determine maximum separation within the range of the sizes of the variables analyzed.

The resulting regression equation is shown below,

$$Y_{\max} = 6.9664 - (0.0329*A) + (3.545*B) + (0.1817*C) - (0.3123*D) - (0.6259*E) \\ - (0.0433*F) - (0.4521*B^2) + (0.0072*C^2) + (0.0618*E^2) - (0.0795*B*C) \\ - (0.0555*B*E) - (0.0329*C*E)$$

where the variables are as follows:

$Y_{\max}$ = Maximum separation (0.001")

A = Die size (31" to 39")

B = Circular post diameter (3" to 5")

C = Cover die thickness (6" to 8")

D = Insert thickness (3" to 5")

E = Ejector die thickness (6" to 8")

F = Proud Inserts (0 to 0.006")

The R^2 for this regression analysis was 0.981.

Design cases

Case	DS	d	t1	t3	t2	0
1	39	3	10	5.0	7	0
2	39	5	10	3.0	10	0.006
3	39	3	8	3.5	6	0.006
4	39	5	7	5.0	8	0
5	39	3	7	5.0	9	0.006
6	39	3	6	3.0	8	0
7	39	5	10	4.0	9	0.003
8	39	5	10	5.0	10	0
9	39	5	10	4.5	6	0.006
10	39	3	10	3.0	8	0.006
11	39	5	9	3.0	7	0
12	39	3	7	5.0	7	0.003
13	39	5	7	3.0	9	0.006
14	39	5	6	4.0	6	0
15	39	4	6	3.0	6	0.006

DS: Die Size

d: Diameter of circular post

t1: Cover die thickness

t2: Ejector die thickness

t3: Insert thickness

0: Amount by which inserts stick out

All dimensions are in inches

Figure 6.7 Cases analyzed

Figure 6.8 Maximum separation for each case

Figure 6.9a shows the interaction between cover die thickness and ejector die thickness with respect to the maximum separation at the parting plane. The figure indicates that the rigidity of the die structure is increased as the thickness of the cover and ejector die is increased. The ejector die is always under compression and is actually stiffer than the cover die even though the ejector die is the moving element. This is due to the cover platen bending around the die and the fact that the cover platen is supported at the corners by the tie bars. As a consequence, die deflections benefit more from increasing the cover die thickness than from the ejector die. However, the combined effect of both the variables is even more significant as can be seen in the figure. The variation in the separation of the dies can be around 0.004" when the thickness is increased from 6" to 10".

Figure 6.9b shows the interaction between cover die thickness and insert thickness. As can be seen the maximum separation is less sensitive to the insert thickness than to the cover die thickness. The insert is bolted to the die structure and hence its effect is dependent on the thickness of the die as compared to its individual thickness. Figure 6.10a shows the response surface for the ejector die thickness and insert thickness. The variation in the maximum separation is about 0.001" for the variation of insert thickness from 3" to 5".

The fact that increasing the insert thickness decreases the maximum separation is somewhat surprising. This is likely a numerical artifact of the way the experiment was carried out. In all cases a minimum die shoe thickness of 2" was kept behind the insert. Hence a thicker insert always accompanied a thicker die. This interaction will be explored more fully in future work.

Figure 6.10b shows the response surface for the cover die thicknesses and die sizes. The maximum parting plane separation is relatively insensitive to the die size. As the die size is increased from 31" x 31" (square) to 39" x 39" (square) the separation decreases by only about 0.001". One of the plausible reasons for the small variation was the small range in which the die size was varied.

Figure 6.11a shows the response surface for the cover die thickness and circular post diameter. It is seen that the maximum separation is insensitive to the size of the circular post until the size is increased to 5". The separation reduces by 0.0015" when the size of the circular post is increased from 4" to 5". The non linear effect is due to the area-diameter relationship.

Figure 6.11b shows the response surface for the cover die thicknesses and the proud inserts. As discussed in chapter 4, the effect of proud inserts is dominant during the initial cycles and its effect decreases as the dies warm up. The effect of higher preload obtained by using a proud insert decreases as the dies warm up and as the contact area at the parting plane decreases. Results indicate that the contact area at the parting plane is the same for the flush inserts and the proud inserts as the dies warm up. The response surface indicates that the proud inserts do not have a large effect on the maximum separation.

Figure 6.9 Cover-ejector and cover-insert interactions

Figure 6.10 Ejector die-insert and cover die-die size interactions

Figure 6.11 Cover die-post diameter and cover die proud insert interactions

A main effects display for each of the variables is shown in Figure 6.12. As expected from the response surfaces, the cover die thickness has the largest effect of all the variables. On a percentage basis, proud inserts and circular post diameter have relatively little contribution. Similarly, the effect of die size is small as long as the die covers at least 50% of the platen. This may or may not be true for dies covering less than 50% of the platen.

Figure 6.13 shows the comparison of the total effect of all the variables on the maximum separation. The total effect of small, medium and large values for all the variables on the maximum separation was compared. It can be seen that the reduction in maximum separation is around 12% when switched from case 1 to case 2, whereas the maximum separation reduces by 33% when switched from case 1 to case 3.

Figures 6.15, 6.16 and 6.17 show the x-y plots of different design variables. There is no new information in these plots. They represent slices through the response surfaces and provide the same information in a simpler form. Figure 6.16 in particular shows that proud inserts have little effect. The importance of cover die thickness is clear from several of the plots.

6.2.4 What-if cases

This section describes a few additional scenarios based on the same model. The following cases were analyzed:

- Comparison of the best design case with the presence and absence of circular post
- Comparison of the best design case on a 1000 ton and 700 ton machine
- Comparison of the best design case with and without the preload on circular post
- Comparison of the best design case on a 1000 ton machine with 800 ton locking force

Figure 6.17 shows the comparison of the different cases. The absence of circular post causes large bending of the ejector die resulting in more separation. The separation increases by around 20% when the circular post is removed. 1000 tons of locking force in a 1000 ton machine decreases the maximum separation by about 38%, the largest decrease seen in the analysis. This decrease is due to increase in the locking force and the thicker platens used in the machine that increase the stiffness of the machine. Note that using 800 tons of locking force in the 1000 ton machine reduces the maximum separation by about 12.5% suggesting that both the extra clamping force and the extra stiffness contribute. On the other hand, using a 700 ton machine results in a 52% increase in the maximum separation.

The preload on the circular post does not seem to have big effect on the maximum separation. The difference in the maximum separation is less than 5% between the cases.

A = Die size

B = Circular post diameter

C = Cover die thickness

D = Insert thickness

E = Ejector die thickness

F = Proud inserts

Figure 6.12 Main effects of the factors

Cases	Size	Die size	Circular post diameter	Cover die thickness	Insert thickness	Ejector die thickness	Proud inserts
Case 1	Small	31"	3"	6"	3"	6"	0
Case 2	Medium	35"	4"	8"	4"	8"	0.003"
Case 3	Large	39"	5"	10"	5"	10"	0.006"

Figure 6.13 Comparison of max separation with factors all set to small, middle and max values

Figure 6.14 x-y plots of max separation for cover die thickness and die size

Figure 6.15 x-y plots of max separation for insert thickness and proud inserts

Figure 6.16 x-y plots of max separation for circular post diameter

Figure 6.17 What if? Cases

6.3 *Effect of Machine on Maximum Separation*

A similar parametric study was conducted by (Choudhury 1997) to establish the effects of some key machine variables on maximum separation at the parting plane. The parameters that were varied, as shown in Figure 6.18, are:

- Tie bar diameter (DTB)
- Platen thickness (PT)
- Die position (DP)

An experimental array was designed for studying the effect of the above variables and their interaction on maximum separation. The tie bar diameter (A) was varied from 6.5 to 8.0 in. with 6.75 in. as the nominal value. The platen thickness (B) was varied from 9.00 to 16.5 in. with 11.75 in. as the nominal value. The die position was varied from 0 to 12.5 in. with 6.25 as the nominal value. When the die position is at 6.25 in., the die is centered with respect to the platen. When the die position is at zero, the die is aligned to the top tie bar center. When the die position is at 12.5 in., the die is aligned to the bottom tie bar center. Each variable had 5 levels. The die used for this study was the nominal case from the experiment discussed in the previous section.

The fractional factorial design cases and the values for different variables are shown in Table 6.2. A FE model was built for each case described in Table 6.2. The model and boundary conditions used for this set of simulations is as shown in Figure 6.19 except that the sizes vary depending on the case under consideration.

For each case the magnitude of the maximum separation at the parting plane was recorded at the point of application of intensification pressure. These values were used to create the response surface of the interactions between tie bar diameter and die position and also between platen thickness and die position.

Figure 6.20 shows the response surfaces. When the die is shifted up from the center the maximum separation increases, but when the die is shifted below the center there is much less variation in maximum separation. This is due to the fact that, although the die is placed at the center of the platen, the center of gravity of the casting is about 1.5 inch above the center of the platen. As a result when the die is shifted up from the center, more pressure is exerted on the top half than the bottom half. But when the die is shifted to the bottom half, the pressure distribution becomes more uniform and the maximum separation is less.

When the die is aligned to the top tie bar center, the maximum separation decreases with increase in tie bar diameter, but it seems to be insensitive when the die is aligned to the bottom tie bar center.

The maximum separation decreases with an increase in platen thickness when the die is shifted from the center of the platen, but the percentage decrease in maximum separation is higher when the die is shifted up from the center of the platen.

Factor		low	high
A.	diameter tie bar (DTB)	5.50	8.00
B.	platen thickness (PT)	9.00	14.50
C.	die position (DP)	0.00	12.50

Figure 6.18 Variables considered in the machine study

$\Pi_{11-3}(5)$ array

	A	B	C
1	1	5	5
2	5	1	5
3	5	5	1
4	5	5	5
5	5	3	3
6	3	5	3
7	3	3	5
8	3	3	3
9	4	1	1
10	1	4	1
11	1	1	4

inches			
	A	B	C
1	5.50	14.50	12.50
2	8.00	9.00	12.50
3	8.00	14.50	0.00
4	8.00	14.50	12.50
5	8.00	11.75	6.25
6	6.75	14.50	6.25
7	6.75	11.75	12.50
8	6.75	11.75	6.25
9	7.38	9.00	0.00
10	5.50	13.13	0.00
11	5.50	9.00	9.38

Table 6.2: Cases for parametric study

Figure 6.19 Schematic of machine, description of boundary conditions

Figures 6.22, 6.23 and 6.24 show the cover side contact pressure between the die and the platen. The die is positioned at the center in Figure 6.22, above the center in Figure 6.23 and below the center in Figure 6.24. In all cases, a is at lock up, b at intensification. The dark areas correspond to zero contact pressure meaning that the platen is not supporting the die at these points. It can be seen from these figures that the contact pressure distribution is fairly symmetric when the die placed at the center at the instant of clamping. The contact pressure is higher at the bottom when the die is shifted up and the contact pressure is higher at the top when the die is shifted below the center of the platen.

At the point of intensification, the top portion of the die opens up when the die is aligned to the top tie bar center. The bottom portion of the die opens up when the die is aligned to the bottom tie bar center. In all cases there is a patch of stress concentration near the box cavity.

The maximum separation decreases with increase in platen thickness and tie bar diameter. This means an increase in stiffness will result in a reduction of maximum separation at the parting plane. It is also clear that loading the machine in an off-center manner can result in very large separations.

6.4 Summary

The results produced are limited, but do present a consistent picture. Solid, uniform support for the die cavity is necessary to minimize separation. For a given machine, cover die thickness and ejector die thicknesses are the most important variables assuming that the die covers at least 50% of the platen. Additional clamping force in a stiffer machine is even more significant. Running a die in a smaller machine results in very significant increases in separation.

The results presented in the two design guideline case studies suggest that proper sizing of the machine and proper loading of the die are critical factors in controlling die deflections. The magnitudes of die separation predicted with the die placed off-center are considerably higher than anything predicted with the experimental array. Separation cannot be eliminated, but clearly it can be made worse by poor use of the machine.

Figure 6.21 Contact pressure between cover die and cover platen, die centered. a) at clamp, b) at intensification

(a)

(b)

Figure 6.22 Contact pressure between cover die and cover platen, die above center. a) at clamp, b) at intensification

Figure 6.23 Contact pressure between cover die and cover platen, die below center. a) at clamp, b) at intensification

7. CONCLUSIONS AND FUTURE WORK

7.1 *Summary of results*

Predicting the mechanical behavior of the die is one step in the journey toward predicting the final part shape with sufficient accuracy that the size and shape of the die cavity can be defined during design accounting for all of the significant mechanical and thermal loads. Modeling at this level will eventually replace the simple shrink factors that are used to adjust the cavity shape. The ability to compute what might be called 3-D shrink factors will enable the die casting process to hold much tighter dimensional tolerances. The work to date is one step along the path toward this goal.

For results to be of practical use, deflections must be computed with a resolution of fractions of a millimeter in models whose size is in the order of a meter. Because of the wide range of conditions, a very important task in this work was that of sorting out the interaction between the different loads, and clarifying what loads are relevant.

A modeling scheme was developed using ABAQUS, a commercial FEM system. The die responses predicted using this modeling method are consistent with what is commonly observed in the field. The modeling procedure was tested on a limited basis against field data.

The use of this procedure opens up the possibility of developing design procedures and design guidelines that account for and minimize die deflections. Work along these lines continues with funding provided by follow-on projects.

The procedures that have been described are general enough to be applied to any die geometry. Details about the scales of resolution of models and results have been addressed. Finally, a general understanding of the response of a die casting die and the importance of the die casting machine has been introduced.

7.2 *Conclusions*

A characterization of the loads upon the tooling in a typical die casting operation has been provided. The results of this work indicate that:

- Momentum and hydrodynamic loads are negligible under typical casting conditions. Only poorly designed die structures or uncharacteristic filling conditions would make these loads significant from the point of view of die deflections.
- Cavity pressures can be modeled by hydrostatic pressure with a magnitude equal to the intensification pressure. Field data confirms that this assumption holds except in those cases in which the application of intensification pressure is delayed beyond the point at which the gate solidifies (ie, beyond the point where the intensification is of value.)
- Injection pressure may result in a dynamic response in the die structure under very specific circumstances. The pressure surges may result in a transient

separation of the die resulting in flash, but the force causing this separation is very short-lived.

- Determining whether or not the die will flash under dynamic conditions can be predicted by scaling the static pressures by roughly 125%. This calculation will predict the peak dynamic separation.
- Dynamic loading will not be significant if deceleration mechanisms are used.
- Generally that maximum sustained deflections will be reached at intensification.
- Heat released during fill may produce deviations from instant fill conditions that can be significant. However, the contributions to die deflections are generally of the same magnitude as other sources of noise, such as manufacturing and assembly tolerances.
- Pressure and thermally induced deflections do not necessarily offset each other's effects. The implication is that the assumption that dies "tighten up" at operating temperature will not necessarily result in favorable lock up stresses.
- In a typical die casting operation, quasi-steady state conditions result when the die has reached a certain periodicity in its bulk temperature. A very large number of cycles (100 or more) are needed to reach quasi-steady state conditions.
- Models that do not account for the machine will significantly underestimate die deflections. Platens, tie bars and toggles at a minimum must be included in deflection models in some manner.

The response of the die casting die can be characterized as follows:

- The stiffness of the die/machine combination ultimately determines the magnitude of the maximum deflections. This stiffness is a function of the design of the die, design of the machine, temperature of the bulk of the die, clamping force, and placement of the die on the machine.
- The characteristic shape of the shutoff pressure, i.e. the pressure that is built up at the surfaces where die halves meet, is affected by the temperature patterns within the die, the location of the toggles on the ejector, and the die cavity shape, the pressures used, the clamping force, and the position of the die in the machine..
- Die design should be driven by the need to reduce deflections at the moment of intensification. Reducing the effects of the pressure spike through the modification of the structure of the die is not likely to succeed in practice. The pressure spike phenomena must be addressed through. plunger deceleration.
- The stiffness of the die and machine ultimately determines the magnitude of the maximum deflections. This stiffness is a function of the design of the die, die bulk temperature, clamping force, and position of the die in the machine.

- The characteristics of the contact pressures or the shutoff pressure at the parting plane is affected by the temperature patterns within the die, clamping force, and position of the die in the machine.

Preliminary work on design guidelines has led to the following recommendations:

- Maximum separation at the parting plane is a suitable performance parameter to use to analyze different die designs. Out of the several performance parameters considered for evaluating die designs, maximum separation was found to be the most sensitive to changes in the die design.
- Proud inserts have some effect on the preload and on maximum separation during the initial cycles, but their effect decreases as the number of cycles increases. This suggests that the insert placement per se is not a useful strategy for controlling die separation and flash.
- Cover die thickness and the ejector die thickness seem to have a large effect on maximum separation. In the case studied, increasing the thickness by about 40% reduced the maximum separation by 0.003".
- Die size (percentage of the platen covered by the die) seems to have less effect on the deflections in the range considered, namely 50% and higher.
- As would be expected, support posts behind the ejector die are important. The results confirm the common sense opinion that the more support the better. Preloading the circular post does not have much effect on parting plane separation.
- The insert thickness does seem to have some effect on the maximum separation when compared to the die size and the size of the circular post. Increasing the insert thickness increases its stiffness resulting in less maximum separation. This result is somewhat suspect and is the subject of additional investigation.
- The effect of using a larger 1000 machine with a die designed for a 800 ton machine decreases the maximum separation very significantly, 38% in the case studied, due to increase in the locking force and thicker platens used in the machine. The effect of using a smaller 700 ton machine increases the maximum separation by 51% for similar reasons.
- The thickness of the cover platen has a significant effect on parting plane separation.
- Proper positioning of the die on the machine platens is extremely important. Poor placement of the die can produce nearly a 100% increase in parting plane separation.

The overall picture that arises from this work is that support for the die cavity is critical. The needed support comes from a properly designed die properly placed in adequately sized machine. Due to the complex shapes present in many die casting die cavities, it is impossible to prevent non-uniform thermal growth, but it appears that it may be possible

to compensate for it. Additional work is needed to reach this goal, but modeling should ultimately meet this objective.

This work was aimed towards predicting die deflections. At this point, the foundations for carrying out such simulations have been laid out and tested against a limited amount of experimental data. The work on design guidelines is incomplete and will require additional work to make it practical and useful to die casters and die designers.

7.3 Future Work

This work has prepared the foundation for design guidelines, but significant work remains to be done. As pointed out in chapter 6, the relationships between the machine platens and the die need to be explored further. All indications at present suggest that the machine dominates the interaction, but the exact relationship between die, machine, and part dimensions is still only partly known.

The spring/platen model is a good first step toward a practical model, but several enhancements are needed. An additional set of spring elements representing the machine toggles are required if the model is to predict the growth in machine loading with heat build up in the die.

Also, the current model spring/platen model cannot account for twisting of the platens. This phenomenon is very likely to be present in dies in which slides are not completely balanced. The shape of the die cavity may also create unbalanced loads. Complex parting surfaces will also create unbalanced loads. In order to account for these cases, some method of accounting for the restraining forces from the tie bars and from the foundation must be developed and added to the model.

The modeling scheme that was introduced in this work is also suitable for diagnosis. For example, given a condition in which the die flashes, the phenomena that causes it can be traced back to either a gap that appears in the parting plane at lock up, deflection due to intensification pressure or pressure spike phenomena. Corrective measures can be taken based on the results of this analysis.

For the purposes of design, the capability of predicting conditions that result in pressure spikes would be desirable. A study in which the plunger deceleration characteristics are correlated with the characteristics of the die (specifically, the runner-feeder-venting system) could provide some insights. Clearly, there is a need for more information before accurate predictions can be made.

Finally, it is important to emphasize that the ultimate goal of any research in this field is improving the quality of the die casting process. The results of the current work can be used to reduce operational problems such as flashing. A natural extension of this work is the prediction of the final dimensions of the casting. The current work makes only a small contribution in this direction: it offers the possibility of predicting the dimensions of the cavity.

Several of these objectives are currently being addressed projects currently underway.

APPENDIX A BASIC FLUID FLOW EQUATIONS

A flow problem is defined by the following set of equations:

Momentum Equations

$$\rho \left(\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = - \frac{\partial p}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \dots\dots\dots (A.1(a))$$

$$\rho \left(\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = - \frac{\partial p}{\partial y} + \mu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \dots\dots\dots (A.1(b))$$

$$\rho \left(\frac{\partial w}{\partial t} + u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = - \frac{\partial p}{\partial z} + \mu \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \dots\dots\dots (A.1(c))$$

Mass Conservation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \dots\dots\dots (A.2)$$

Subject to the boundary conditions $u=v=w=0$ where the fluid is in contact with the die walls, plus

$$\frac{\zeta}{r_1 + r_2} = p \dots\dots\dots (A.3)$$

where ζ = surface tension and $r_{1,2}$ are the principal radii of curvature of the free surface. This boundary condition is needed to establish the shape of the fill front. Additionally, in the region corresponding to the location of the biscuit:

$$p(t) = \text{prescribed}$$

This system is coupled with the energy equation within the fluid

$$\rho c_p \left(\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = k \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \dots\dots\dots (A.4)$$

and the energy equation within the die

$$\rho c_p \frac{\partial T}{\partial t} = k \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \dots\dots\dots (A.5)$$

(internal heat generation and other second order terms have been neglected), the energy balance at the interface solid-liquid within the fluid

$$\rho H \frac{\partial S}{\partial t} = k \frac{\partial}{\partial n} (T_l - T_s) \dots\dots\dots (A.6)$$

initial conditions

$$T_i = T_l \text{ where } T_i > T_{\text{liquidus}}$$

and boundary conditions

$$\frac{\partial T}{\partial n} = 0 \quad \text{at liquid - air and solid - air interfaces}$$

and

$$-k \frac{dT}{dn} = h(T_d - T_l) \quad \text{at liquid/die interface}$$

This system constitutes the complete problem statement during die fill. Equations (A.1) are commonly known as the Navier Stokes Equations⁸. Except for a few cases, no

⁸ Equations A.1 are a particular form of the Navier Stokes Equation. The general form contains terms that account for effects of gravity and fluid compressibility.

analytical solution exists for these equations. In the case of die casting, the presence of jet or atomized flow requires that the boundaries of the fluid front in the vicinity of obstacles be computed as part of the solution, thus complicating the problem immensely.

APPENDIX B THERMAL SIMULATIONS FOR THE BOX PART

The thermal loads must be computed in order to carry out deflection/stress analysis. Figure B1 shows the location of the cooling lines for the box part suggested by NADCA Computer Modeling Task Group. The boundary conditions used for the thermal analysis are also shown in Figure B1. Heat transfer analysis was performed for this configuration of cooling lines. The thermal loads were computed by first simulating the die geometry in MAGMAsoft (MAGMA Foundry Technologies, Inc.) and using the outputs of this simulation to estimate the loads upon the die.

The simplest approach to calculate die temperatures consists of carrying out a heat transfer analysis for which the magnitude of the heat from solidification is prescribed. Numerous experimental studies (ABAQUS 1996) have shown that heat released by the casting in to the die proceeds approximately at an exponentially decaying rate, irrespective of location. In his work, (Padiyar and Miller 1994) took advantage of this behavior to model the thermal load upon the die. His basic assumption was that heat escaping from the part flows in a direction perpendicular to the cavity surface only, and that the part is very thin.

The total heat released by the casting in question (W/sq.m) for the above part configuration is given by the equation

$$Q_{\text{total}} = m_{\text{casting}} C_{p,l} (T_{\text{injection}} - T_{\text{liquidus}}) + m_{\text{casting}} L + m_{\text{casting}} C_{p,s} (T_{\text{solidus}} - T_{\text{ejection}}) \dots \text{(B1)}$$

where

$T_{\text{injection}}$ - bulk temperature of the cast material

T_{liquidus} - liquidus temperature of the cast material

T_{solidus} - solidus temperature of the cast material

T_{ejection} - bulk ejection temperature of the casting

m_{casting} - mass of the casting (= density * volume) (in Kg)

$C_{p,l}$ - specific heat of liquid metal (in J/kg °C)

$C_{p,s}$ - specific heat of solid metal (in J/kg °C)

L - Latent heat of solidification (in J/kg)

Thermal fields in the die were obtained by using an assumption that the heat flux over the cavity surface decreases exponentially with time. It was assumed that the heat flux at the time of ejection has decayed to around 1/100 of the initial value of the heat flux at the time of injection. Figure B2 shows the assumed heat flux variation. Based on this assumption, the heat flux is calculated using the following equations.

$$q'(t) = K.10^{mt} \quad \text{(B2)}$$

where K,m are constants which are empirically determined

q' = heat flux as a function of time t. (W/sq.m)

Also,

$$A_{\text{die}} \int_{t_{\text{inject}}}^{t_{\text{eject}}} q'(t) dt = Q_{\text{total}} \quad (\text{B3})$$

where

A_{die} = area of the die cavity surface in contact with the die casting (sq.m)

Q_{total} = is the total heat transferred from the casting to the die surface (J)

It is assumed that

$$q'(t_{\text{eject}}) = \frac{1}{X} q'(t_{\text{inject}}) \text{ where } X \text{ lies in the range of } 50 \text{ to } 100 \quad (\text{B4})$$

Given a total heat release (Q_{total}) for a given region, constants K and m in Equation (4.2) can be computed based on the constraints imposed by equations (B3) and (B4).

$$K = \frac{Q_{\text{total}} \cdot m \cdot \log_e(10)}{A_{\text{die}} \cdot (10^{mt_{\text{eject}}} - 10^{mt_{\text{inject}}})} \quad (\text{B5})$$

$$m = \frac{-\log_{10}(X)}{(t_{\text{eject}} - t_{\text{inject}})} \quad (\text{B6})$$

In essence, the accuracy of the approximation depends upon the values used for K and m. The trick of course is being able to match the part temperature at ejection (T_e) after a time ($t_{\text{eject}} - t_{\text{inject}}$).

In calculating the heat fluxes, the casting was divided into several thermal zones and the heat flux was calculated by applying equation (2) to each thermal zone. Each zone had different ejection temperatures owing to the different geometry and location in the casting. In each of these zones different values for T_{ejection} were assumed. The ejection temperatures in each of this zone were obtained by solidification simulation of the casting in MAGMA. For each of these thermal regions K and m were calculated, based on the formulas given above. Table B1 shows the K and m values of the different thermal zones used in the simulations.

The heat removed from the die cavity surface during lubricant spraying, was modeled as a negative heat flux and applied on free element faces in the finite element model. It was assumed that 15% of the heat added during solidification, was removed during this process. The heat transfer coefficients and the process conditions used in the simulation are shown in Table B2.

The time interval between the material injection and the ejection of the part was 35 seconds and the total cycle time was 55 seconds. The percentage solidification obtained using this time period was 98.63%. Several simulations were carried out in order to

arrive at the above value of cycle time. Figure B3 shows the simulation results obtained for a cycle time of 35 seconds and Figure B4 shows the simulation results for a cycle time of 55 seconds. Based on this result a cycle time of 55 seconds was selected for the above part.

Table B1 Thermal zones and respective K and m values

Thermal zone	During material solidification		During coolant spray	
	K (W / m ²)	m (s ⁻¹)	K (W / m ²)	m (s ⁻¹)
Overflow	9.779E+05	-2/35	-1.708E+06	-0.6666
Box	9.870E+05	-2/35	-1.727E+06	-0.6666
Gate	1.197E+06	-2/35	-2.095E+06	-0.6666
Runner	1.827E+06	-2/35	-3.197E+06	-0.6666
Biscuit	2.314E+06	-2/35	-4.049E+06	-0.6666

Table B2 Assumed casting conditions

Process parameters	Process conditions
Casting Material	Al 380
Melt Temperature	670 °C
Maximum Cavity Pressure	70 MPa (10,000 psi)
Cycle Time	55 seconds
Clamping Force	7250 kN (800 tons)
Ambient temperature	20 °C
Heat transfer coefficient (steel - air)	20 J/ °K
Heat transfer coefficient (steel - steel)	2000 J/ °K
Heat transfer coefficient (steel-cooling	10000 J/ °K
Temperature of platens	80 °C
Water temperature	40 °C

Based on the temperature results, the total amount of heat released by the part can be estimated, from which values can be assigned for K and m in different regions. These

values were used in a heat transfer simulation with ABAQUS. Thermal patterns were then compared with the results of MAGMA, and the values of K and m can be modified if significant differences exist. This process is repeated until temperature fields in both simulations are similar. Table B.3 shows the die casting cycle used in the analysis.

Table B3 Assumed steps in a single casting cycle

Time (sec)	Step
0-1	Dies are clamped
1-36	Heat and pressure applied to the Cavity
36-47	Die opens, part is knocked out
47-50	Lubricant spray
50-55	Dies are clamped

Figure B5 and B6 show the comparison of the results obtained from MAGMA and ABAQUS for $X=50$, 100 and 125 at the different sections of the die. It is seen that for $X=100$ the temperature distribution seems to match the temperature distribution obtained from MAGMA. The value of X determines how fast is the heat added to the die as a function of time. A higher value of X would add heat rapidly during the initial cycle time and then decrease rapidly at the later stage of the cycle while a lower value of X would add heat at a constant rate during the whole cycle.

Figure B1: Location of the cooling lines and the boundary conditions used in the thermal simulation.

a) Heat flux variation with time

Figure B2: Figure showing the variation of the heat flux with time.

Figure B3 Temperature pattern from MAGMA at end of solidification for cycle time of 35 seconds

Figure B4 Temperature patterns at end of solidification for cycle time of 55 seconds

Figure B5: Comparison of the temperature distribution for the box and and gate sections of die from MAGMA and ABAQUS.

Figure B6: Comparison of the temperature distribution for the runner and biscuit sections of the die from MAGMA and ABAQUS.

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