# Multi-Stage Plunger Deceleration System

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## ABSTRACT

Die flashing is an ongoing challenge for many die cast operations. One significant factor that contributes to this production problem is the shot end deceleration that applies a large force on the die and the machine locking system. A variety of solutions are currently used to reduce this factor in production, machine low impact hydraulic settings, SoftSHOT design methodologies, and other industry proprietary methods. At Albany-Chicago we have developed a damping mechanism built into the overflow runner system that dissipates a portion of the excessive force that leads to flash generation, the resulting instantaneous loss of cavity pressure, and reduced casting properties.

## INTRODUCTION

This research project is a collaboration between Flow Science, Inc. (FSI) and Albany-Chicago, Co. (ACC). We build on the work presented this year at the World Foundry Congress in Chennai, India [Palekar A., Starobin A., Reikher A., 2008]).

The objective of this paper is to begin the comparison of a multi-stage plunger deceleration system developed at Albany-Chicago, Co. with the computational results obtained at Flow Science, Inc. The ACC design combines overflow and gas ventilation system runners, effectively decelerating moving parts of the die cast machine at the end of cavity fill, as well as allowing utilization of a portion of kinetic energy of the moving part of the shot system to produce effect of early intensification. The FSI computations help to identify essential features of the design and provide detailed information about the end-of-fill pressure and velocity transients. The paper is in two sections; one area of research focuses on the computational models used to analyze and predict the metal flow and some aspects of machine behavior. The second section reflects some of the early testing associated with the numerical predictions and the potential for flash reduction. While the testing is not yet complete, the results from the first trials are promising.

We used a challenging 19 lb, A380 casting as our initial test case of the new overflow design and measurement methodology. The casting requirements include pressure integrity both in oil and coolant channels, and superior overall mechanical properties to withstand harsh use in the field. The casting system was optimized for flatness and distortion using ANSYS® [ANSYS, Inc., Canonsburg, PA] analysis tools and numerous FLOW-3D® [Flow Science, Inc, 2007] gate simulations. The casting design achieved first run dimensional, automation, and run-at-rate success, some of which is associated with the new overflow back pressure system.

The role of pressure in high pressure die casting process is well known and is described in prior work [Mickowski J.R. *et al*, 1993, Herman E.A., 1988 and Savage G. *et al*, 2001]. The importance of impact and intensification pressure on the quality of casting is both documented [Savage G. *et al*, 2001] and relied upon in the industry to produce quality results. It was shown that the number of rejected castings decreases with an increase in both intensification and impact pressure. Impact pressure has positive effect on the quality of the casting, but it is only effective until the machine locking forces are exceeded by metal pressure forces.

Increase in intensification pressure improves quality, however the peak useful pressure is ultimately limited by early partial freezing of the gates. With the ACC back pressure system we can allow maximum overflow gate area, maximum vent areas, and fast fill times, while minimizing the die flashing forces.

## METHODS

A typical die casting machine monitoring system is insufficient in that it does not report directly in-cavity metal pressure. In order to gauge the effectiveness of our overflow designs a dynamic load cell (see Fig. 1) was installed under the ejector pin in the main runner area. True-Track 2020 monitoring system was used to record the remaining machine parameters: first and second stage velocity and hydraulic pressure. A series of 3 separate die trials were conducted. Plunger diameter, percentage of fill, and metal temperature remained the same during all trials. In-cavity pressure was measured continuously. The results of the measurements presented were for the duration of the cycle time.

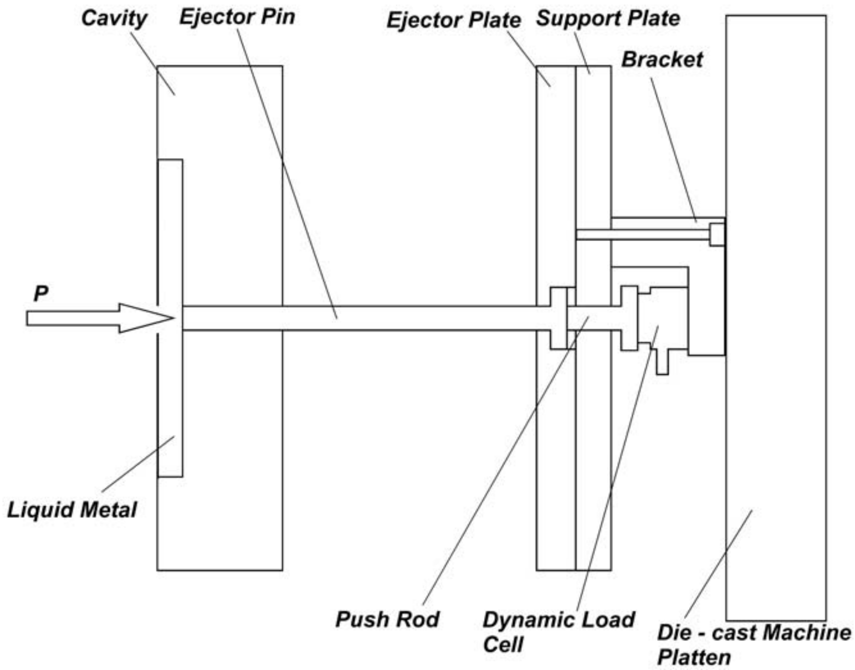


***Figure 1. Schematic of Experiment Setup***

## GATING, RUNNER AND OVERFLOW DESIGN PROCEDURE

Die cast process parameters were calculated using [Reikher A, Barkhudarov M, (2007)] die cast process calculator. FLOW-3D® [Flow Science, Inc., 2007] was used to simulate alternate gating designs. The final runner design was chosen based on the best flow pattern, which resulted in a coherent flow front, filling the casting from the runner end (critical area of the casting) to the overflow end. The overflows and ventilation system were positioned at the last filled area which allowed maximum evacuation of the gas from the die cast die cavity (see Fig. 3 and 4).

Subsequent thermal analyses were conducted. Results of thermal analyses were used to position cooling channels. ANSYS was then used to determine the final casting shape after ejection out of die cast die.

The production design is shown in Figure 2. The real estate area required for the runner system will depends on size of the casting, mass of the moving parts of the die cast machine, fast shot velocity, and threshold set for maximum allowable impact pressure. The only added requirement is the decelerator which can be placed between casting and outer edge of the die.

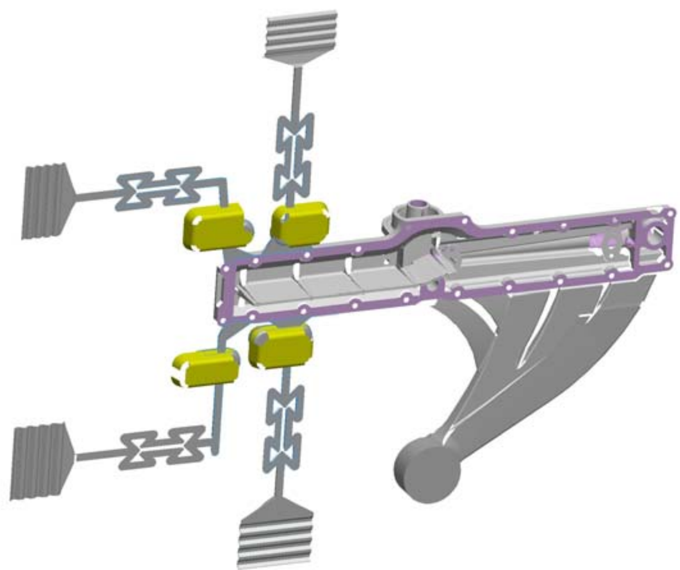

*Figure 2. Casting with Multi-Stage Deceleration System*

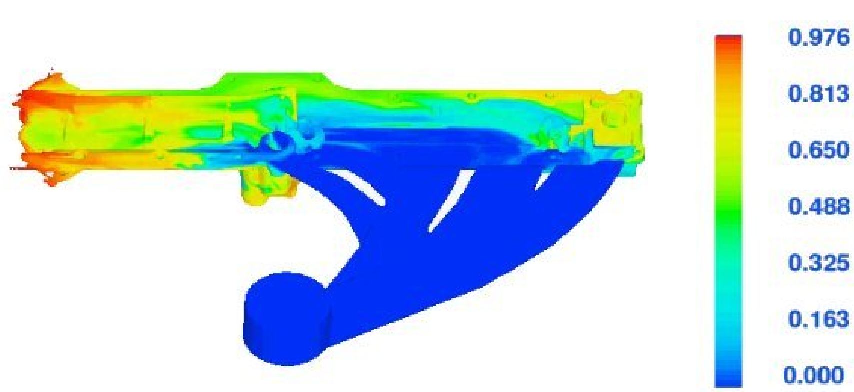


*Figure 3. Flow analysis results from a prescribed motion simulation*
*(Scale: Volume fraction of Entrained Air)*

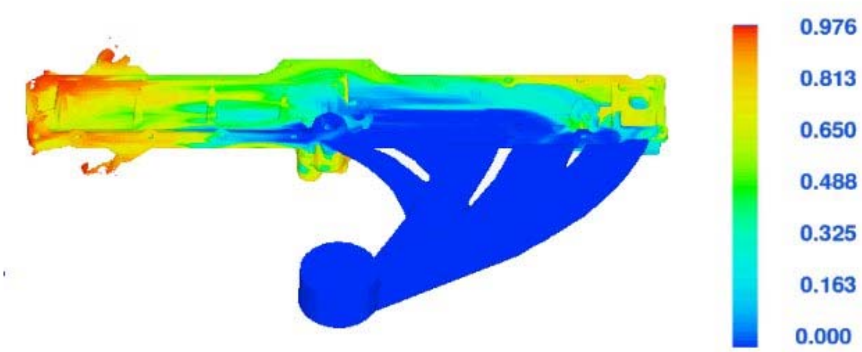


*Figure 4. Flow analysis results from a prescribed motion simulation*
*(Scale: Volume fraction of Entrained Air)*

## MEASUREMENT RESULTS

A dynamic load cell was used to register pressure changes at various stages of the die cast process. Figure 5 shows results of the hydraulic pressure recorded by True-Track monitoring system. Figure 6 shows results of the measurements of the cavity pressure. As it can be seen on Fig. 5 impact hydraulic pressure is 1411 PSI (9.73 mPa). Shot cylinder bore diameter is 8.5'(215.6 mm), plunger diameter is 4.75' (120.65). Based on these parameters, expected cavity pressure at impact is 4518 PSI(31 mPa). Measured pressure at impact 4200 PSI (29 mPa) amounts to 6% error. Expected static cavity pressure is 4060 PSI (28 mPa). Measured static cavity pressure is 3770 PSI (26 mPa). Difference between measured and expected static cavity pressure is 7%. Measured pressure when metal reaches the gate is 426 PSI (2.94 mPa) Fig. 7 shows the results of cavity pressure calculations at the same point in time. There is 5% error between numerical and measured results of the cavity pressure. The calculation of Fig. 7 was done using True-Track velocity as input (see the next section for simulation methodology).

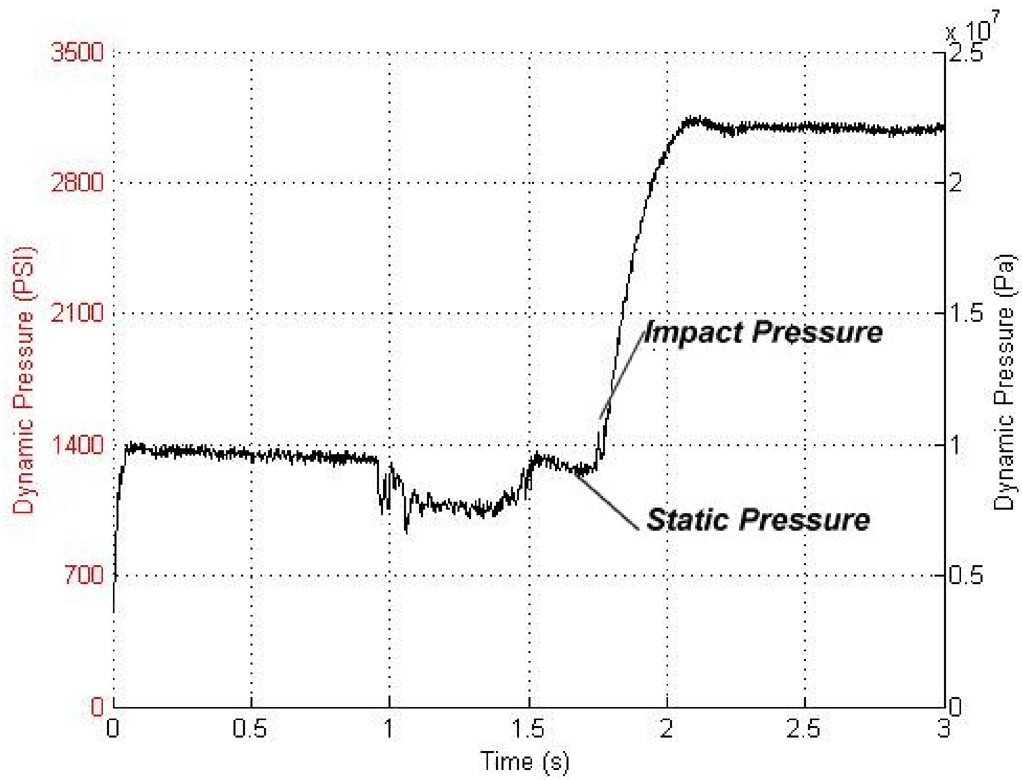


***Figure 5. Plunger driving pressure (True-Track monitoring data)***

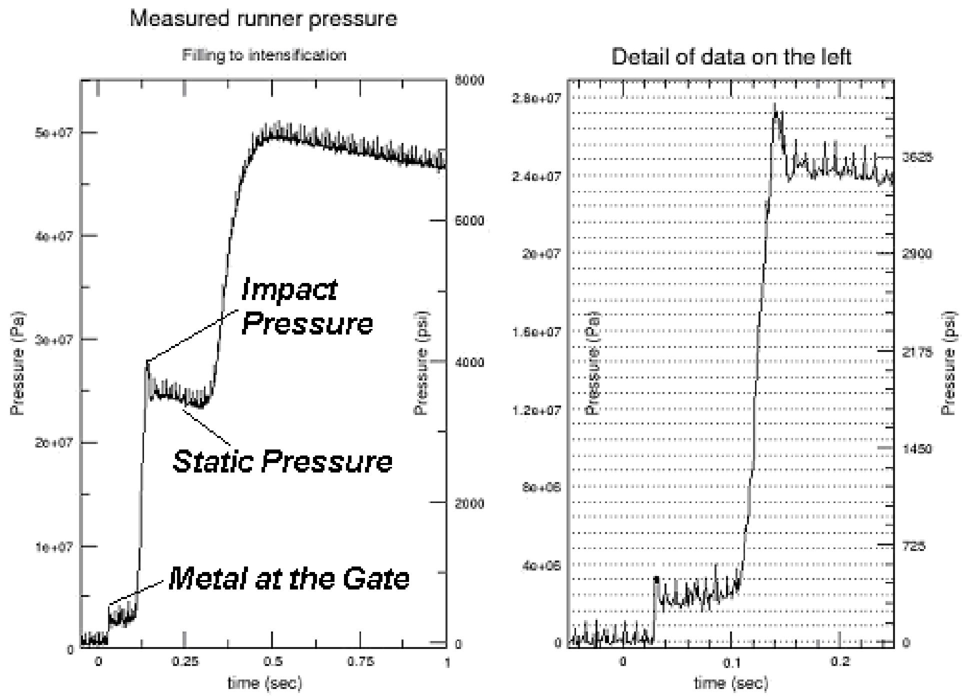


*Figure 6. Measured Cavity Pressure for design 3. Left panel: Filling, Impact and Intensification. Right panel: zoom of the data in the left panel around the fast stage and the end-of-fill impact*

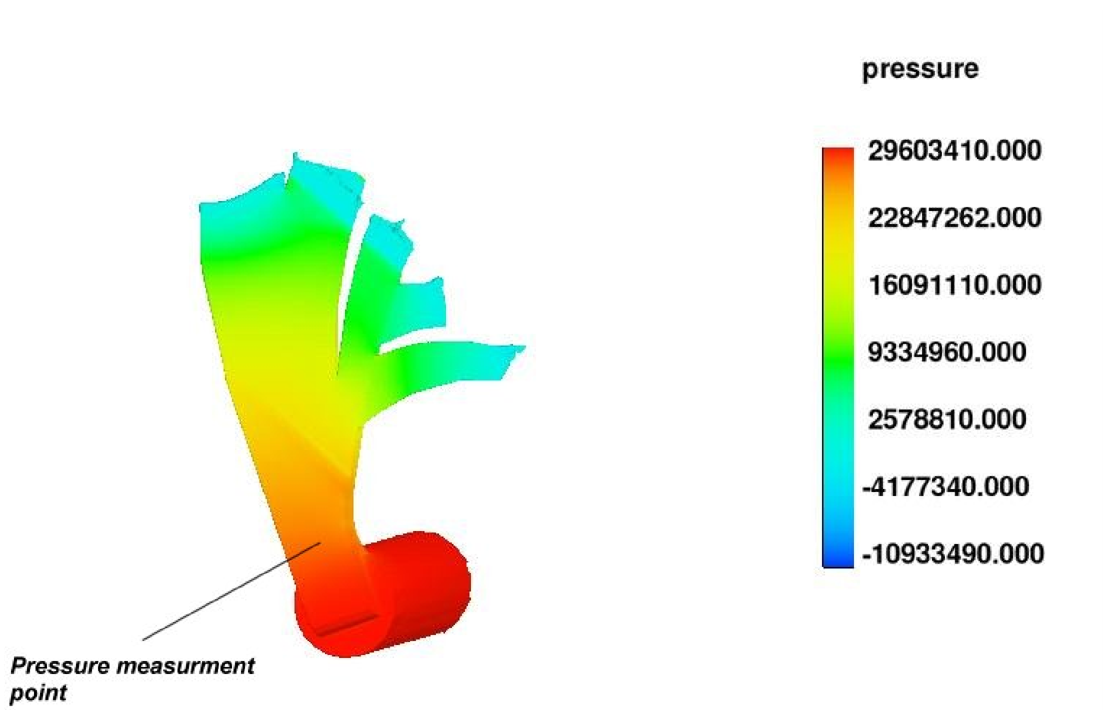


*Figure 7. Simulated cavity pressure at runner full with prescribed plunger motion (Pa)*

## COUPLED-MOTION MODEL FOR THE PLUNGER

At the design stage one operates with “dry” slow and fast shot velocities. The real shot profile will differ from the assumed one in several ways. First, the transition time from slow to fast shot is limited by the available power and inertia in the hydraulic system. The fast shot speed will typically decay during fill due to accumulator pressure drop, frictional losses, and the rise of cavity back pressure. Finally, as the main cavity fills the plunger, all the components of the hydraulic system will begin to decelerate and the in-cavity pressure will ascend to the static pressure as the flow resistance increases abruptly and the metal begins to flow though smaller overflow gate area.

Attempts have been made [Xue et al, 2005] to specify more realistic shot profiles which would include a velocity ramp-down at the end of fill. However, in general the beginning of the ramp and its slope cannot be estimated well at the design stage, which leads to possibly significant errors in the prediction of the peak total pressure and its dynamic component (The dynamic pressure is defined as the excess pressure above the static [Savage G. *et al*, 2001]). The slope of the velocity ramp at the end of fill is controlled by a number of factors: partial pre-fill, if any, of the overflows, total inertia at the end of main cavity fill left in the system and overflow gate areas and volumes. Since there are typically several overflows cut into the die, there is uncertainty as to which subset will control the final deceleration behavior. If good guesses can be made for the fill sequence, then the SoftSHOT methodology [Branden J., *et al*, 2002] appears to be adequate and is being used to size overflow gate areas and volumes.

At FSI we have developed a simple model for the hydraulic system by allowing a fully coupled metal plunger motion in our computations of the die fill. The model uses real plunger mass and allows for position dependent hydraulic driving force. As in our past developments, we utilize the Fractional Area-Volume Obstacle Representation (FAVOR™) technique to describe the object geometry in fixed rectangular meshes by means of area fractions ($A_f$) and volume fractions ($V_f$) [Hirt, C. W., Sicilian, J. M., 1985 and FLOW-3D® Manual v9.2, Flow Science, Inc., 2007]. In each computational control volume, $V_f$ is defined as the ratio of the volume open to fluid to the total cell volume, and $A_f$ is defined at each of the six faces as the ratio of the respective open area to the total area.

A fixed-mesh method based on the FAVOR™ technique is extended to general moving objects (GMO) to model fully-coupled motion of solid bodies in fluids [Wei G., 2005]. At each time step, $A_f$ and $V_f$ are updated in accordance with the object’s motion. This fixed-mesh GMO method has advantages over the moving and deforming mesh methods because it treats complex moving objects very efficiently and conveniently. The motion of each moving object is not restricted in its complexity. A physically acceptable treatment of collisions between objects is also possible.

The effect of the plunger on the liquid metal is facilitated by introducing appropriate source terms into the equations of fluid mass, momentum and energy transport [Wei, G., 2005, Barkhudarov M., Wei G., 2006]. The source terms are a function of the plunger velocity, while the latter is a function of pressure and viscous shear forces of the metal on the plunger. At each time step, the equations of rigid body and fluid motion are solved in a coupled fashion. Effects of hydraulic force (pressure and shear stress), gravitational force (a necessary but likely trivial contribution given the magnitude of the other forces involved) and control force on the plunger’s motion are considered. Locations and orientations of the plunger are tracked in space, and area and volume fractions are updated accordingly.

General motion of a rigid body can be divided into a translation along with a reference point and a rotation. In the present work we restrict ourselves to modeling only irrotational and linear motion of the shot sleeve plunger along the sleeve’s main axis. The total force acting on the plunger consists of a prescribed control force as well as pressure and viscous forces acting from the metal. The control force is a sum of the plunger side machine hydraulic force and possible frictional forces between the plunger and the shot sleeve walls. Typical shot monitoring systems report only time dependent accumulator pressure which can differ considerably from the actual hydraulic system side plunger pressure. The two are only equal under static conditions, otherwise pressure loss across the high speed flow control valve must be accounted for.

For the simulations described below we take a simplified view of the control force, by noticing that for steady state plunger motion during the fast shot the driving force must be exactly balanced by the back pressure of the metal. As was described in the earlier work [Palekar A. *et al.*, 2008] this leads to a simple expression for the control force, $f$, in terms of the “dry” fast shot speed, $V_{FS}$, main gate area, $A_g$, plunger area, $Ap$, and metal density, $\rho$ :

$$f = \frac{\rho}{2} A_p \left( \frac{A_p}{A_g} V_{fs} \right)^2$$

In earlier simulations, [Palekar *et al.*, 2008], of a simple vertical one-gate runner system the above expression predicted driving force vs. fast shot speed dependence for a range of driving forces to 5%. The “dry” fast shot speed chosen for the fill of the design of Figure 2 is 3.65 m/s. The plunger diameter and area are 0.12 m and 1.13e-2 $m^2$ respectively. The geometric gate area in the design is 1.04e-3 $m^2$ and the metal density is 2500 $kg/m^3$. This gives for the driving force 24.5 kN.

In the current study we have an additional benefit of a measured in-cavity pressure (Figure 6). We can use this extra information as an additional check on the driving force. The difference between the metal pressure force on the plunger and the rate of change of inertia of the plunger should give a time-dependent driving force for a particular shot. The plunger acceleration is obtained by differentiating the velocity history available from True-Track 2020. The mass of the plunger used for conversion and in the simulations of the next section is 177 kg.

The driving force obtained from measured in-cavity pressure and the True-Track 2020 information is plotted in Figure 8. The data covers an interval from the end of the slow shot through about the middle of the fast shot. The force reaches 20 kN early on and continues to increase slowly to 26 kN during the fill. This is in good agreement with the force value based only on “dry” fast shot velocity and design geometry.

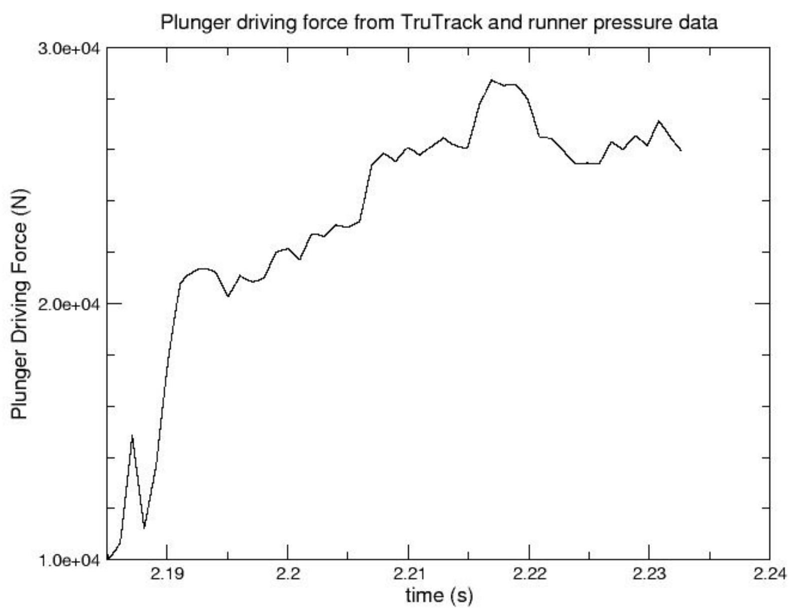


***Figure 8. Net driving force on the plunger computed from in-cavity pressure data of Figure 6 and the True-Track2020 record of plunger velocity history***

## COMPUTATIONAL RESULTS FOR THE PRODUCTION DESIGN AND THREE ALTERNATE DESIGNS

Using the model and input parameters described in the previous section we evaluated four overflow designs for the runner system and the part of Figure 2. Design 3 is the production design with the total overflow gate area about three quarters of the main gate area. Design 4 employs a 1:1 gating ratio and also uses multi-stage overflow. Designs 1 and 2 have gating ratios of 0.5 and 1 respectively and have a simpler one-stage overflow that does not include the thin long overflow runner.

- All the computations were performed with version 9.2.1 of commercially available CFD package FLOW-3D® [Flow Science, Inc., 2007] on a 2 four channel processor Intel machine with 8Gb of memory running under 64 bit Windows

2003 server addition. The structured one-block mesh contained approximately 21 million cells with a 3.8 mm x 3.8 mm resolution in the plane normal to the shot sleeve and a variable 2.5 mm to 5.1 mm resolution in the direction of plunger motion. The only active cells in the computation were those open to flow which numbered only 0.9 million. A typical compute time utilizing both processors was 6 hours with the main cavity filling.

- The flow was assumed turbulent and a renormalization group k-ε turbulent viscosity model with a mixing length of 1 mm was used to model turbulent viscous losses. Metal cooling and solidification was ignored in these simulations and the simulation interval started at the onset of the fast shot. The initial plunger and metal velocity was specified to be 0.5 m/s which was the set "dry" slow shot velocity in production. Plunger position and metal fill level at the beginning of the simulation were deduced from known metal volume and known duration of the slow shot. The metal level is such that at the onset of fast shot the runner is full and the metal is just inside the main cavity

The computed plunger velocity and runner pressures for the production design (design 3) are shown in Figure 9. Also, plotted in figure 9 is the measured TrueTrack shot profile from the onset of the fast shot to the end of fill. The agreement between velocity histories is good. The acceleration time, peak fast shot velocity and deceleration times are captured well in a constant driving force simulation.

The computed runner pressure is in satisfactory agreement with the measured pressure during fill which ranges from 1.9 MPa early on to 2.4 MPa right before impact (Fig. 6, Right panel). Part of the experimental pressure creep is due to the slow increase in the static pressure throughout the fill not captured in the simulation.

Both in the simulation and in the actual shot the part fills between 85 and 90 ms. During this time the pressure rapidly increases to allow for the previously achieved high flow rate now maintained through smaller overflow gates. The peak simulated pressure recorded is recorded in this interval and is $\sim$ 7.5 MPa. The pressure in excess of steady fast stage pressure is ~ 5.5 MPa. The complete filling of the part is delayed by 45-50 ms. During this time the plunger is decelerated to below 0.2 m/s and the final impact pressure is only $\sim$ 5 MPa. Unlike in the description of the impact process given by Savage *et al.* [Savage G., *et al*, 2001] the peak impact pressure is generated right as the metal gets to the overflows and not at the end of fill. In design 1 and 2 inertia in the metal and plunger at the end of fill is appreciable and the description of the system by Savage *et al* as a spring with some composite stiffness $K_V$ is appropriate and can yield an estimate of final impact pressure spike and the period of the subsequent ringing.

One clear discrepancy between the computed pressure in Fig. 9 and the measured pressure in Fig. 6 is the absence of the static pressure ramp at the end of fill in the computational results. This is natural since the driving force in the simulation is fixed. In the machine the pressure increases to the static value as the metal and plunger come to rest.

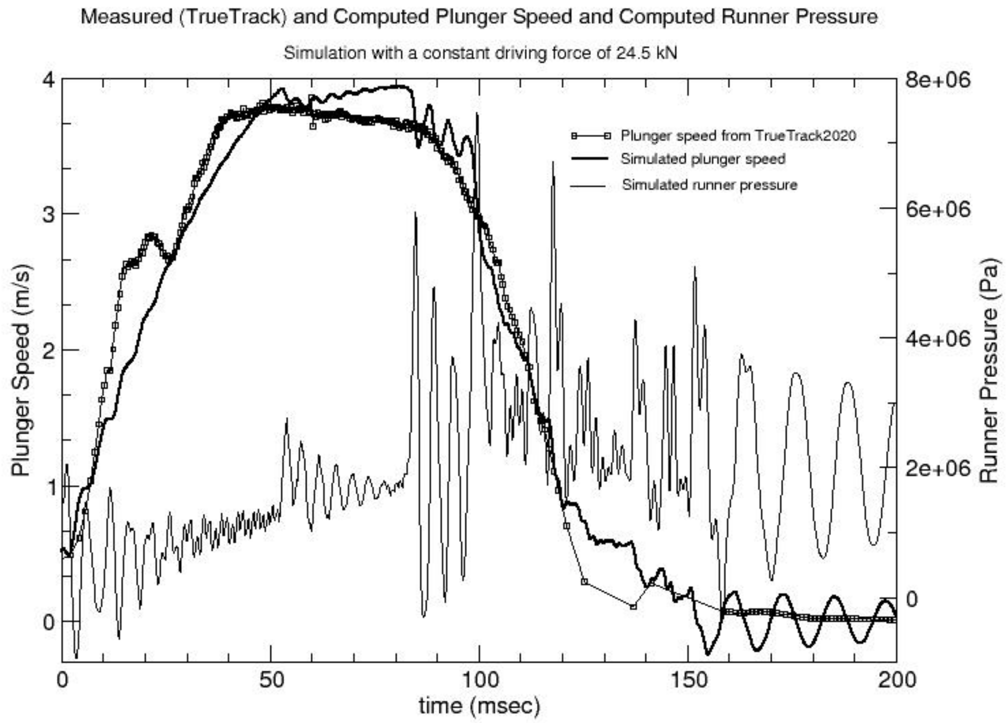


***Figure 9. Measured and computed plunger velocity history and computed runner pressure profile for design of Figure 2 with a prescribed driving force of 24.5 kN***

***Table 1 Computed impact pressures and deceleration times with a constant driving force of 24.5 kN for four overflow designs***

| Design # | Ratio of overflow to main gate areas | Multi-Stage Overflow | Peak impact pressure (MPa) | Plunger velocity at end-of-fill (m/s) |
|---|---|---|---|---|
| 1 | 0.5 | No | 9 | 0.4 |
| 2 | 1 | No | 36 | 2.9 |
| 3 | 0.75 | Yes | 7.6 | 0.14 |
| 4 | 1 | Yes | 9 | 0.2 |

The simulation results from production design and the three other designs are summarized in Table 1. Though further reduction of peak impact pressure is possible if the overflow gate area is slightly increased, the production design appears near optimal. The next best design is Design 4 with a multi-stage overflow system. The peak computed pressure in this case is 9 MPa and the residual velocity at the end of fill is only 0.2 m/s. The worst design is Design 2 with the 1:1 overflow to main gate area ratio. For Design 2, peak impact pressure is 36 MPa and little deceleration of the plunger is achieved during overflow fill.

## CONCLUSIONS AND FUTURE WORK

The aim of this study was to verify effectiveness of the Multi – Stage Deceleration system, and to compare numerical analysis with measured values of the cavity pressure. Production runs show the ability of the system to create back pressure at the end of the fill process. The Multi – Stage Deceleration system allows the use of a portion of the dissipated kinetic energy of the moving parts of the die cast machine. We claim this elevates cavity pressure and metal to flow through the cavity during the late phase filling moment. This metal, under increased pressure, moves through the overflows, and the ventilation system runners until they are completely filled. It also allows the impact pressure to remain below critical flash generation levels. The simulations of the cavity filling using a new coupled motion numerical model for the plunger show that the production design is near optimal. Both the velocity history and in-cavity pressure are predicted with a fixed driving force as the only input parameter. Satisfactory correlations between numerical and measured results of the cavity pressure are demonstrated.

As the new overflow design was tested on a new die, with a newly optimized gate, runner, and overflow system, there was not a previous test case that could have allowed photographic or measurement comparison of levels of flash between the new and the previous die. The current die is showing very little flashing but we cannot, at this time prove with photos (before and after) that the low flash levels are caused only by the decelerator presence. We plan a future test removing only the overflow decelerator shape, replacing it with a straight-through channel, to measure the impact of the decelerator alone, on the fill and flash conditions. While we are confident there is a strong correlation, further proof will emerge in our 2008 tests. Additionally, further testing is underway with other casting shapes to determine if the results can be repeated with varying casting volumes. We will focus on larger area castings (200-400 $in^2$) requiring static pressures above 5,000 PSI (34.5 mPa). This year, we will also run a series of trials on this initial die application to optimize the overflow mass to further fine tune the pressure applied to casting areas closest to the overflow gates. We anticipate an equation set to allow for rapid optimized design, narrowing parameters and minimizing simulation runs to reduce design expense.

## ACKNOWLEDGMENTS

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