Mathematical modelling of air evacuation in die casting process via CASTvac and other venting devices

L. H. Wang*

Air venting in the die casting process has been an issue since its inception. A mathematical model that predicts the venting efficiency via different devices, including a new technology known as CASTvac, is presented. This model offers an advantage over other models reported in the literature in that it includes air leakage through die parting faces and the rates of volume compression due to a rapid change of plunger speed. The model has been verified and has predicted that (a) a majority of air is removed during the first stage of injection, in the case of atmospheric venting either using a conventional chill vent or CASTvac; (b) the duration of vacuum application is critical for a simple vacuum system due to die leakage; and (c) CASTvac used as a vent can achieve nearly the same evacuation efficiency as the best simple vacuum system.

Keywords: Die casting, Air venting, Air evacuation, Mathematical modelling, Chill vent, Duct flow

List of symbols

- $A$: cross-sectional area of a duct, m$^2$
- $D_h$: hydraulic diameter, m
- $f$: friction factor
- $k$: specific heat ratio of air, 1-40 for 20°C and 1-39 for 200°C
- $L$: vent length, m
- $m$: air mass at time $t$, kg
  where $m = \bar{m} N$
  and $\bar{m}$ is the mole weight, 0.028964 kg mol$^{-1}$
  for air
  $N$: the number of mole
- $M$: Mach number
- $p$: air pressure at the entrance of duct, Pa
- $p_B$: back pressure at the exit of duct, Pa
- $R$: gas constant
  where $R = \frac{\bar{R}}{\bar{m}}$
  and $\bar{R}$ is the universal gas constant,
  8.3143 J mol$^{-1}$ K$^{-1}$
- $t$: time, s
- $t_s$: time for vacuum stop, s
- $T$: air temperature, K
- $u$: air velocity at the entrance of duct, m s$^{-1}$
- $V$: air volume, m$^3$
- $\Delta m$: mass exchange of air at time step $\Delta t$, kg
- $\rho$: air density, kg m$^{-3}$

Subscripts

- 0: stagnation properties
- 1: properties at duct inlet
- 2: properties at duct exit
- $i$, $i+1$: $i$th and $(i+1)$th time step

Superscript

- $*$: critical or sonic properties

Introduction

High pressure die casting is a versatile process for producing a wide range of metal products from automobile engine blocks to toys. In the process, molten metal is forced under high pressure into a cavity formed by metal moulds called dies. Since a highly turbulent flow occurs during the metal injection phase, air entrapment is unavoidable. The air entrapped in the castings is the main source of defects such as porosity, blisters and misruns. Efforts to minimise the air entrapment are made from the commencement of die design by optimising the runner, gate and vent designs and also during daily operation of the die by varying the operational parameters.

Vacuum technology is adopted in some foundries to assist in the removal of air from the cavity. Due to extra cost added to production and the unreliability of current mechanical valve system, a majority of die casting machines are operated without a vacuum. Air removal in this case relies on atmospheric venting either through vents (typically 0.1–0.2 mm gaps formed by cutting grooves around the cavity on the die parting faces) or through a chill vent (a zigzag venting path typically 0.5–1.0 mm of gap formed by two metal blocks). This conventional chill vent is sometimes connected to a vacuum system to enhance air removal. The advantage of using a chill vent as a vacuum valve is its robustness since no moving parts are involved in the operation. However, its small venting area restricts the venting...
efficiency. Further increase of its venting area by increasing the width of the chill face will take up the die projected area. The author and colleagues1 have described in the previous papers,2–4 the author concentrates here on those aspects of the model where improvement is felt to be necessary.

A few models for the calculation of air venting in the die casting process have been reported in the literature. A valuable critique and comments on these models have been well detailed in the papers by Bar-Meir et al.2,3 and Nouri-Borujerdi et al.4 Most of the models, including the latest ones,2–4 treated the air flow in the vent as a Fanno flow (adiabatic flow in a duct with friction). This was the best approximation to this application to date since an accurate calculation of the heat exchange between the die and the air has been elusive. Rather than repeating the literature review, which has been well described in the previous papers,2–4 the author concentrates here on those aspects of the model where improvement is felt to be necessary.

All of the previous models assumed a constant injection velocity by ignoring the first stage of metal injection. In reality a die casting machine runs at least two stages of injection. The volume compression at the first stage is greater than that at the second stage when the fill ratio in the shot sleeve is often less than 50%. This model uses a real injection trace, which includes the two stages of velocity and the transition velocity between the two stages, as the inlet boundary condition. It is also interesting to know the amount of air vented out at the first stage from a practical operation point of view.

During vacuum assisted casting, air leakage through the gap between the parting faces of the die is unavoidable. In the case of atmospheric venting, the gap between the dies acts as an additional vent. A second vent is introduced in this model to simulate this problem. It acts as an air leak for the case of vacuum and an additional vent during atmospheric venting.

It is interesting to note that all of the models published have lacked verification. In this paper, two sets of experimental data have been used to validate the model. Two critical parameters used in the model, the friction factor and the gap size between the die parting faces, have been determined with the aid of this data.

Model

A typical die casting process is schematically shown in Fig. 1. In the process, the molten metal is poured into the shot sleeve and injected into the cavity with a plunger driven by a hydraulic system. The atmospheric venting or vacuum evacuation takes effect only after the pouring hole is covered by the moving piston.

It is generally believed that the air flow is restricted by the resistance at the entrance of the chill vent or vacuum valve. The flow resistances at other locations (in the shot sleeve and the runner, at the gates, in the cavity and the vent runner) are assumed to be negligible. Based on this assumption, the injection process can be simplified as the one shown in Fig. 2. That is, a piston compresses an air chamber and the air is vented out through a duct (duct 1). The air chamber represents the total volume of the partially filled shot sleeve, the cavity and the runners, while duct 1 represents the vents, chill vent or vacuum channel depending on the cases applied. Duct 2 is used to simulate the gaps between the die parting faces as mentioned above.

The air mass in the chamber is determined by the air pressure, temperature and volume according to the ideal gas law

$$m = \frac{pV}{RT}$$  \hspace{1cm} (1)

for the time step $\Delta t$ the volume is compressed by $\Delta V$ and the mass exchange between the chamber and the environment is $\Delta m$. On differentiating, equation (1) becomes

$$\frac{\Delta m}{\Delta t} = \frac{1}{RT} \left( \frac{\partial V}{\partial t} + p \frac{\Delta V}{\Delta t} - \frac{V}{T} \frac{\Delta T}{\Delta t} \right)$$  \hspace{1cm} (2)

equation (2) states that the mass change depends on the
changes of pressure, volume and the temperature of air in the chamber, and vice versa. It is noted that the volume change can be readily calculated from the operational parameters (first and second speeds of the plunger advancement and the shot sleeve dimension). However, an accurate calculation of the air temperature in the cavity is very difficult, since it depends not only on the mechanical work of compressing the air, but more importantly on the heat exchange of air with the dies. The die temperature is influenced by many factors, for instance, die spray, water quench, steam generation and air blow, etc. For simplicity, all of the previous models assumed the air compression in the cavity as an adiabatic process based on the argument that the cavity filling takes only a few hundredths of a second. This is not quite true if it is considered that the first stage of injection takes a few seconds and the dies have a high heat capacity due to their large mass. By considering these factors, this model has assumed an isothermal process for the first stage of injection and an adiabatic process for the second stage when the temperature of air in the chamber is calculated.

Further the process investigated is an open system. In principle the air temperature is affected by the mass exchange as described in equation (2). To simplify this problem, the compression at each time interval is divided into two steps in this model, namely an isolated compression followed by an isothermal mass exchange. It should be pointed out that this assumption has only been applied for the temperature calculation. The calculation of air pressure has been fully coupled with the mass exchange and will be discussed in the following section.

Based on the above assumption, at the first stage the temperature of air in the chamber is constant (ΔT=0) as it is regarded as an isothermal process. When the plunger starts to move at a high speed in the second stage, an adiabatic process is applied. At this stage the change of air temperature is determined solely by the change of volume

\[
\frac{T_{i+1}}{T_i} = \left( \frac{V_i}{V_{i-1}} \right)^{k-1} \tag{3}
\]

when the mass exchange Ω at the time interval Δt is expressed as the air flow through the ducts, the following equation is obtained

\[
\Delta m = \rho u A \Delta t = p_0 M_1 \left( \frac{k}{RT} \right)^{1/2} \left( 1 + \frac{k-1}{2} M_i^2 \right)^{-(k+1)/(2(k-1))} A \Delta t \tag{4}
\]

where \( u = M_1 (kRT_i)^{1/2} \), \( T_i = T \left( 1 + \frac{k-1}{2} M_i^2 \right)^{-1} \), and other symbols are defined in the symbol list.

In the model, the unknowns (ρ and m) are determined by equations (2) and (4), while the air temperature in the chamber (T) and the remaining air volume (V) can be predetermined by the operational parameters according to the above assumptions. The air flow through the duct, represented by the Mach numbers, is determined by the vent conditions and the air states (ρ and T) across the vents. The Mach number M is calculated as follows.

For an adiabatic flow in a duct with friction, the critical length function is defined as

\[
\frac{fL^*}{D_0} = 1 - \frac{M_i^2}{kM_i^2} + \frac{k+1}{2k} \ln \left( \frac{(k+1)M_i^2}{2(1+(k-1)/2M_i^2)} \right) \tag{5}
\]

when the critical length \( L^* \) is equal to the physical length of the duct \( L \), the Mach number at the exit reaches the maximum (\( M_2 = 1 \)). By the time, the flow is choked and the Mach number at the duct entrance also reaches the maximum, which is defined as the critical entrance Mach number \( M_1^* \) in this paper (\( M_1^* \) is always less than 1 when \( L = 0 \)).

By taking the pressure in the chamber as the stagnation pressure \( p_0 \), the critical pressure (choke pressure) \( p^* \) at the exit can be calculated from the following equations when substituting \( p \) with \( p_1 \) and \( M \) with \( M_1^* \)

\[
\frac{p_0}{p^*} = \left( 1 + \frac{k-1}{2} M_1^* \right)^{k/(k-1)} \tag{6}
\]

\[
\frac{p}{p^*} = \frac{1}{M} \left( 1 + \frac{(k+1)/2}{1+(k-1)/2M_1^2} \right)^{1/2} \tag{7}
\]

the flow is choked when the back pressure at the exit \( p_0 \) is lower than the critical pressure \( p^* \). By the time the Mach number at the duct entrance is equal to \( M_1^* \) as stated before. When the flow is unchoked, the Mach number at the exit \( M_2 \) has to be calculated from the following relationship

\[
\frac{fL^*}{D_0} = \frac{fL^*}{D_0} = fL^*_{M_2} = fL^*_{M_1} - fL^*_{M_1} \tag{8}
\]

where \( fL^*_{D_0} \) and \( fL^*_{D_0} \) are calculated from equation (5) when substituting \( M \) with \( M_2 \) and \( M_1^* \) respectively.

A similar flowchart to the one in the Ref. 5 (p. 821) has been used to calculate \( M_2 \). The ratios of \( p_2/p^* \) and \( p_1/p^* \) in the flowchart can be calculated from equation (7) when \( M \) is replaced with \( M_2 \) and \( M_1^* \). The new \( M_1 \) can be obtained from equation (6) when the ratio of \( p_1/p_0 \) is known.

The algorithm to solve the model for pressure and mass is shown in the flowchart in Fig. 3. It starts from the input of initial condition (IC) and boundary condition (BC). For each time step, it calculates the Mach numbers at the inlets and outlets of the ducts with a guessed chamber pressure, works out the total mass exchange according to equation (4) and then modifies the chamber pressure until convergence is achieved. The iteration goes on for each time step until the time for the shot end is reached. The algorithm for the calculation of Mach numbers (denoted in the box by dashed lines) has been stated above.

**Verifications**

This model has been refined with the data from two scenarios. Part of the objective for this verification as mentioned above was to determine two parameters (the friction factor for air flow in a duct and the gap size for air leakage in a real die) via an inverse calculation.

The first dataset was obtained from a bench test conducted for the measurement of evacuation efficiency of a pair of chill vent. In the test, one end of the chill vent was connected to a 3 L tank (a cavity simulator)
while the other end was to a vacuum tank. The test setup has been detailed in a paper\(^6\) published previously.

Since the system in this test was well sealed and the air leakage was negligible, the air leak area in this model was therefore zero and the only unknown became the friction factor in this case. A trial-and-error approach by varying the friction factor \(f\) in the equations (5) and (8) was employed to fit the pressure curve obtained from the above test. It was found that an \(f\) value of 0.07 gave the best fit to the measured curve. The result has been plotted in Fig. 4. When the Colebrook formula and Moody chart in a standard textbook\(^5\) were used to estimate the friction factor for this case, the \(f\) value was approximately in the range of 0.01 to 0.025. A large \(f\) value obtained from this calculation could be due to the resistance at the connections with abrupt expansions and contractions (they are not considered in the Colebrook formula and Moody chart).

Bar-Meir\(^2,3\) estimated that the dimensionless resistance coefficient \(fL/Dh\) for the venting in a typical die-casting process was 3 to 7 or greater. Substituting the dimension of the chill vent used in the test (\(L=0.15\) m, \(Dh=2\times100\times0.5/100=1\) mm) together with the \(f\) value derived above gave the resistance coefficient a value of 10.5, which agreed with Bar-Meir’s estimation. It should be noted that this calculation was based on an input of the hydraulic diameter \(Dh\). However it was found in our previous work\(^9\) that the chill vent had an evacuation efficiency equivalent to a tube with a diameter of 4.93 mm. This equivalent diameter was thus used in this calculation rather than the hydraulic diameter as normally used in the textbook.

The second test was conducted in a toolroom of a die-casting plant. In this test, a real die was installed on a 200 t press and connected to a portable vacuum unit as shown in Fig. 5. A hole left by the shot sleeve was plugged using an aluminium biscuit. Pressure sensors were installed in the plug and in the vacuum line. When the measurement took place, a compression force of 200 tonnes was applied to simulate the shut-off condition as in a real die-casting machine.

A commercial mechanical vacuum valve with an opening area of 80 mm\(^2\) was used in this test. The opening area was nearly the same size as the vent runner. The vent runner together with the vacuum valve was therefore treated as a duct and the total length of 0.5 m was used in this calculation. Using the friction factor 0.07 derived above and the measured pressure from sensor 2 as the boundary condition at the exit of the duct, an effective leak area of approximately 10 mm\(^2\) was obtained by taking the same trial and errors approach as for the friction factor above. The predicted pressure together with the measured pressures has been plotted in Fig. 6.

The obtained effective leak area has been further verified with Huebscher’s equation\(^7\) which converts a rectangular geometry to an equivalent circular diameter \(d_e\):

\[
d_e = 1.30(ab)^{0.25}/(a + b)^{0.25}
\]

where \(a\) and \(b\) are the lengths of major side and minor side respectively.

The die used for this test had a disk shaped cavity with an equivalent diameter of 220 mm. Assuming that the die had a uniform gap of 0.1 mm when the two halves were shut-off and substituting this value together with the perimeter of the cavity (690 mm) into Huebscher’s equation, it gave an equivalent diameter of 3.58 mm and thus an area of 10.05 mm\(^2\). This value is consistent with the value obtained above. This evaluation indicates that the assumption of a same friction factor of the leak as the vent is reasonable.

In summary, this model fitted the measurement well when a suitable friction factor in the first case and a leak area in a real die were chosen. It should be noted that these parameters are greatly dependent on the configuration and condition of a real die applied. The friction factor 0.07 was derived from the result of a bench test. The hose size (one inch) and the connections were
similar to those of a vacuum system in a real die investigated, but the vent runner and the gate were not included in the test rig. When these factors are considered for a real die, either a slightly larger \( f \) value should be used or the duct length should be increased as it has been done in the second test. Both can achieve the same outcome. Second, the die leak area of 10 mm\(^2\) represents only this particular die under the test conditions used. In reality a new die can be shut off better than a worn die. In production, aluminium flash and die lube can affect the die shut-off condition too. Nevertheless this is the first time in the literature that a quantitative estimation of the die leak area has been given, which can be valuable for modelling, die design and practical operation.

**Predictions**

This verified model has been used to simulate a few venting methods for comparison. All of the simulations were based on the same die conditions (approximate that of the die heavily used for CASTvac trials in industry). The simulated die had volumes of cavity 0-6 L, biscuit 0-3 L and both the runner and vacuum runner 0-3 L. The piston, which had a diameter of 85 mm, was simulated to advance in two stages with the first and second speeds of 0-25 and 3-0 m s\(^{-1}\) respectively. The simulated plunger stroke was 0-4 m from covering the pouring hole to the shot end. The simulation started from the time when the pouring hole was covered \((t=0)\). The plunger was changed from the first speed to the second speed at \( t=1 \) s and took 0-05 s to complete the transition. When a vacuum was applied in a simulation, the vacuum source with a pressure of 10 kPa was on at \( t=0-2 \) s and the vacuum time (the time from vacuum on to off) varied from 0-4 s to the time till the shot end. A temperature of 200\( ^\circ \)C (an average die temperature) was assigned to the initial temperature of air in the chamber while the ambient air temperature for the simulation of air intake was assumed to be 20\( ^\circ \)C.

The result shown in Fig. 7 has demonstrated the importance of taking into account die leakage in this model. In this simulation, CASTvac was the venting device with vacuum applied. As stated in the introduction of this paper and demonstrated in our previous paper, the venting capacity of CASTvac is 4 times that of a 4-93 mm diameter tube. The hydraulic diameter of CASTvac is therefore twice the tube diameter, namely 9-86 mm. As can be seen in Fig. 7, the difference between the inclusion and exclusion of the die leakage is significant. When the leakage is ignored, the predicted air pressure in the cavity reaches the minimum pressure 10 kPa (vacuum tank pressure) within 0-5 s from the vacuum start. When the leakage is considered, it takes 0-6 s to reach an equilibrium pressure \( \sim 20 \) kPa. By the time the equilibrium pressure is reached, the air evacuation rate through the vacuum valve is nearly equal to that of the air intake from the gap between the die parting faces. When the plunger is changed over to the high speed at \( t=1 \) s, this equilibrium is interrupted due to a rapid compression of the air volume. As a result, the air pressure in the cavity increases rapidly.

It should be added that in this and the rest of the calculations the leak area started with a constant value of 10 mm\(^2\) and was gradually reduced to zero as the cavity volume was proportionally reduced.

It should be emphasised that the air pressure in the cavity does not represent the amount of gas remaining in the cavity. The air mass in the computational domain is therefore used for the evaluation. Furthermore, as can be seen in Fig. 7, even though the air pressure rapidly increases at the second stage of injection, the remaining air mass instead decreases rapidly due to the air venting caused by a fast compression of the air. All simulations were terminated at the time of the cavity full (the vacuum runner not filled yet), since air entrapment after this point was not of concern. As illustrated in Fig. 7, the air masses remaining in the domain at the time of the cavity being full are 0-010 and 0-023 g respectively for the cases of exclusion and inclusion of the die leakage (0-01 g of air represents a volume of 8-3 cm\(^3\) which has a spherical diameter of 25 mm at 20\( ^\circ \)C and one atmosphere).

The following example has shown the application of this model for a system commonly used in the industry which can be called a simple vacuum system. In this system the vacuum line is opened and closed by a valve which is activated by a signal triggered by the plunger position. Using this system an early shut-off will lose the vacuum in the cavity because of air drawing back. Taking account of the activation lag, the valve is normally shut off not later than the change over time
for the sake of safety to prevent the penetration of metal into the vacuum system. Three values of the evacuation time (0.4, 0.6 and 0.8 s) have been selected in the simulation to show its influence on the vacuum level in the cavity. The changes of air mass under these conditions have been plotted in Fig. 8. In the figure, the time from zero to 0.2 s is the leading time normally used in the vacuum operation for the settling down of waves in the shot sleeve. During this period of time, the valve is shut off and the mass reduction as shown in the figure is caused by the air expulsion through the gap between the parting faces as represented by duct 2 in the model. When the vacuum source is introduced at \( t = 0.2 \) s, the air mass decreases rapidly, as does the air pressure, which is not shown. At the end of the evacuation, i.e. the time when the valve is off, the air starts to be drawn back. As a result the air mass in the cavity increases. As shown in the figure, the evacuation time has a significant influence on the final air mass in the chamber. This indicates that the valve shut-off time in a simple vacuum system is critical.

The air evacuation and atmospheric venting by CASTvac and a conventional chill vent have been simulated using this model for a comparison. In the simulation, the chill vent has the same geometry as the one used in the bench test as mentioned before (100 mm in width by 150 mm in height with a gap size of 0.5 mm), while the parameters for CASTvac are the same as those listed in Fig. 7.

The simulation result has been illustrated in Fig. 9. As shown in the figure, when both are used as a vent, the air masses in both cases decrease linearly during the period of low plunger speed at nearly the same rate. This can be explained by the fact that the air has sufficient time to be expelled even though the venting area of a chill vent is small. However, the difference becomes apparent in the period of high plunger speed. The eventual difference of the air mass between the use of CASTvac and the chill vent is 3.55 times (0.098 g with CASTvac compared with 0.348 g with the chill vent). By considering the initial air mass of 1.69 g in the chamber, the air masses remaining in the chamber by the time of the cavity full (not the vacuum runner yet) are 5.8% for CASTvac and 20.6% for the chill vent. It is also noticed that at the end of the first stage \( t = 1 \) s, more than 60% of air (62% for CASTvac and 61% for the chill vent) have been removed.

In the case of both connected to a vacuum, the air mass is reduced significantly faster by the use of CASTvac than that of a conventional chill vent when the vacuum is introduced at \( t = 0.2 \) s as shown in Fig. 9. The removal of air at the first stage is 93% for CASTvac and 71% for a chill vent. Compared with the results for the atmospheric venting as discussed above, the introduction of vacuum for a chill vent can only promote the efficiency by 10% (from 61 to 71%), while CASTvac can do 31%.

These simulation results in Fig. 9 are expressed in Fig. 10 as the final air mass versus different venting methods according to calculation. Parameters for chill vent were \( L = 0.15 \) m, \( D_h = 4.93 \) mm; for CASTvac \( L = 0.5 \) m, \( D_h = 9.86 \) mm; other parameters were \( A_{\text{leak}} = 10 \) mm\(^2\) and \( L_{\text{leak}} = 0.25 \) m, \( p_2 = 10 \) kPa, \( f = 0.07 \), \( T_{\text{die}} = 200 \) °C.
simple vacuum system. CASTvac with vacuum is significantly better than the best simple vacuum system, because it can sustain the evacuation until the cavity is full, while in a simple vacuum system the air can be drawn back after the valve closes.

It should be pointed out that this prediction is based on the assumption that the metal flow in the cavity follows a plug flow pattern, i.e. the air is always ahead of the metal front. In the die casting process, the filling pattern is very complicated and may be different from this ideal case. However, the conclusion is still valid since the comparison is based on the same assumption.

Conclusions

A mathematical model for air venting and evacuation has been presented in this paper and used to simulate the performance of different venting devices.

This model has demonstrated that it is crucial to consider air leakage between the die parting faces, which has not been addressed in previous models, and the duration of vacuum application is critical for a simple vacuum system due to the air drawing back after the valve is shut off before the end of the metal injection.

The following conclusions can be drawn from the prediction of this model.

1. More air is removed at the first stage of metal injection than that at the second stage when either a conventional chill vent or CASTvac is used for the atmospheric venting of air from the cavity.

2. CASTvac when used as a vent is more efficient than a conventional chill vent connected to a vacuum system, and can achieve nearly the same evacuation efficiency as the best available from a simple vacuum system.

Acknowledgements

The author would like to thank the CAST Cooperative Research Centre for funding this work. The data used for the verifications were obtained with the assistance of my colleague Ms M. Gershenzon. Thanks are also given to my colleagues Dr T. Nguyen, Dr C. Davidson and Dr R. O’Donnell for their helpful discussion.

References


