INFLUENCE OF MOLD GEOMETRY ON HEAT TRANSFER, THERMOCOUPLE AND MOLD TEMPERATURES IN THE CONTINUOUS CASTING OF STEEL SLABS

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ABSTRACT

Continuous casting molds are routinely instrumented with thermocouples that give only part of the information needed to characterize mold heat transfer, which is very important to understanding and predicting steel quality. Three-dimensional finite-element models are used to study the relationship between mold geometry and heat transfer and the results are distilled into simple analytical relationships. Specifically, a one-dimensional conduction model is verified that makes use of an effective heat transfer coefficient to account for water cooling of the mold. This model is shown to be increasingly accurate as the ratio of (slot spacing - slot width) to slot depth is small and the ratio of slot width to slot spacing approaches unity. An offset distance is introduced to enable the one-dimensional model to accurately relate thermocouple temperatures to heat flux by accounting for 3-D conduction effects from the complex local geometry. The offset distance depends chiefly on mold geometry and is virtually independent of boundary conditions. For typical molds with thicknesses of 25 to 45 mm, the calculated offset distances range from 4 to 10 mm depending on local slot spacing and depth near the thermocouple. A simple relationship is developed to predict narrowface corner temperatures, based on the distance from the end cooling slot to the narrowface corner. The smaller the distance between the end slot and the corner, the lower the corner temperature. The relationship provides an easy tool to help design narrowface slots to achieve desired hot face temperatures.

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NOMENCLATURE

<u>Symbol</u>	Designation	<u>SI Units</u>
А	= angle of end water slot measured from cold face	[Degrees]
a	= intermediate coefficient used to calculate Nu _D	[-]
b	= intermediate coefficient used to calculate Nu _D	[-]
Cp _{water}	= water heat capacity	$[J kg^{-1} K^{-1}]$
d _{ch}	= water slot depth	[m]
d _m	= distance from straight water slot root to mold hot face	[m]
d _{ml}	= mold thickness	[m]
doffset	= correction distance	[m]
d _{thcpl}	= distance from thermocouple weld point to d_m	[m]
Н	= variable height of end water slot measured from cold face	[m]
h	= heat transfer coefficient	$[W m^{-2} K^{-1}]$
h _{fin}	= equivalent fin heat transfer coefficient	$[W m^{-2} K^{-1}]$
$h_{\rm w}$	= water-side heat transfer coefficient, specified on cooling water slot surfaces	$[W m^{-2} K^{-1}]$
k	= thermal conductivity	$[W m^{-1} K^{-1}]$
k _m	= mold thermal conductivity	$[W m^{-1} K^{-1}]$
k _{thcpl}	= thermocouple thermal conductivity	$[W m^{-1} K^{-1}]$
k _{water}	= water thermal conductivity	$[W m^{-1} K^{-1}]$
L^*	= dimensionless length	[-]
L _c	= distance from end water slot surface to mold hot face corner	[m]
L _{ch}	= straight water slot spacing	[m]
L _{eff}	= effective length	[m]
MN	= minimum temperature in an Ansys isotherm plot	[°C]
MX	= maximum temperature in an Ansys isotherm plot	[°C]
Nu_D	= Nusselt number, dimensionless number comparing the total heat transfer to conductive heat transfer	[-]
Pr	= Prandtl number, dimensionless number comparing momentum diffivity to thermal diffusivity	[-]
q″	= heat flux	$[MW m^{-2}]$
q _{int}	= heat flux at the interface between the mold and mold powder	[MW m ⁻²]
q"hot,cu	= heat flux on hot face of copper mold	$[MW m^{-2}]$
q ["] cold	= heat flux leaving mold cold face	$[MW m^{-2}]$
Re	= Reynolds number, dimensionless number comparing inertial to viscous effects	[-]
T^*	= dimensionless temperature	[°C]
T _c	= mold hot face corner temperature	[°C]
T _{cold}	= cold face temperature	[°C]
T _{film}	= film water temperature, average of T_{cold} and T_{water}	[°C]
T _{hot}	= hot face mold temperature	[°C]

T _{thcpl}	= predicted thermocouple temperature	[°C]
T _{thcpl,3-D}	= thermocouple temperature from a 3-D finite element model	[°C]
T _{water}	= water temperature	[°C]
T _x	= variable through-thickness mold temperature at coordinate x	[°C]
Vwater	= water velocity	$[m s^{-1}]$
W	= variable width of end slot	[m]
Х	= through-thickness mold coordinate, zero at straight water slot root	[m]
μ_{water}	= viscosity of cooling water	[Pa-s]
$\mu_{water,f}$	= water viscosity calculated at film temperature	[Pa-s]
$\rho_{water,f}$	= water density calculated at film temperature	[kg m ⁻³]
CON1D	= one dimensional heat transfer model	
t-t	= section through the mold thickness that passes through the thermocouple weld point	
f-f	= section through the mold thickness that passes through the middle of the fin	
W-W	= section through the mold thickness that passes through the water slot	

1. INTRODUCTION

A sudden change in mold temperature signals a warning to a caster of an impending breakout failure. Mold temperatures, particularly the hot face temperature, are also of interest to casters because of their effect on gap heat transfer, strand surface quality, and the prevention of various defects. Narrowface corner crushing and the resulting fin defects, possibly even sticker breakout failures, are thought to be a result of high, hot face temperatures at the corners of the narrowface.^[11] Therefore, mold temperatures are continuously monitored during the casting process and an abundance of mold temperature data are available from industrial casters. Mold temperatures are also of use to the research community since they can be used to predict heat flux to the mold. This heat flux can then be used to model heat transfer and improve mold design and other casting parameters.

Although it has been demonstrated that three-dimensional calculations are important in capturing mold heat transfer,^{[1],[2],[3],[4]} methods of converting thermocouple temperature to heat flux are often used without regard for the multidimensional thermal behavior around each thermocouple site.^[5] Still others, who recognize this difficulty in converting temperatures to heat flux, present thermocouple temperature profiles which are often interpreted to have the same shape as heat flux profiles.

Therefore, fast, accurate ways to relate thermocouple temperatures to heat flux, estimate narrowface-corner temperature, and optimize mold water slot geometry for desired mold temperature profiles are needed without the need to always resort to 3-D computations.

1.1 **Objectives**

An improved 1-D heat transfer model will be developed through comparison with threedimensional, finite element model results for two commercial molds. To do this, a correction factor, called an offset distance will be developed to account for multidimensional thermal behavior around thermocouple sites. Finally, the enhanced 1-D model will be used to predict mold heat transfer from more accurately interpreted thermocouple temperatures.

Because the narrowface corner temperature plays an important role in known defects, a relationship will be formulated to estimate corner temperatures from simple analytical calculations so that end water slot geometries can be planned to achieve objective temperatures.

1.2 Process Overview

A schematic of the continuous casting process appears in Figure 1.1.^[6] Molten steel is poured from the ladle into the tundish through a submerged entry nozzle and into a water-cooled copper mold. Mold flux powder is shoveled on top of the steel in the mold and serves a lubricant between the mold and the solidifying strand and becomes part of a protective slag layer. The slag layer floats on top of the molten metal acting as a barrier against the oxidizing atmosphere, entrains impurities from the metal, conserves heat, and transfers heat from the strand to the mold.^[7] Once in the mold, the molten metal flows over the water-cooled copper mold faces and solidifies as it progresses downward. The partially solidified strand exits the mold, supported by foot rolls, and continues to solidify as it is cooled by water sprays. Finally, the continuous strand is torch cut at a point beyond the metallurgical length, the point at which the stand is completely solid. The transportable slabs are about two to 20 inches thick, three to six feet wide, and 10 to

50 feet in length. The spray cooling process requires the most space, requiring about 50 yards length from the mold to the torch cut-off station. The caster, made up of the mold and the cooling water containment system, takes up most of a three-story building.

A close-up of the mold region of a slab caster appears in Figure 1.2.^[1] A top view appears in Figure 1.3.^[1] The copper mold is surrounded by a steel water jacket that contains the cooling water. There are four movable faces that make up the slab mold, a pair of widefaces and a pair of narrowfaces. These faces are held together by an external clamping force that exceeds the ferrostatic loading of the molten steel. This additional clamping load is applied as a safety factor and is transmitted across the narrowface, concentrating at the narrowface corners. To aid in the cooling of the mold, cooling water channels, or water slots, are machined into the back of the copper plates. Water is forced from the bottom of the mold to the top and is contained by the steel water jacket which is attached to the mold with bolts. The bolts are exposed to cooling water as they pass through the water jacket, and into a plastic plug in the copper mold. The bolt interrupts the standard straight water slot spacing resulting in additional space between slots near the bolts. An instrumented mold, one that contains thermocouples to monitor mold temperatures, often feeds thermocouples down the axis of mold bolts. The thermocouple passes through a hole in the bolt, through the plastic plug, and into the mold. Some thermocouples have a constantan wire that is welded to the copper mold and others are the self-contained, sheathed type that are inserted into the mold with conductive paste.



Figure 1.1: Continuous slab casting process schematic (from literature)^[6]



Figure 1.2: Close-up of caster (from literature)^[4]



Figure 1.3: Top view of caster (from literature)

2. MODELING HEAT TRANSFER IN CONTINUOUS CASTING MOLDS

Because of its high resistance to heat flow, the interfacial gap between the strand and mold governs heat flux through the mold.^{[8],[9]} Interfacial heat transfer has been the topic of several models based on summing thermal resistances of the molten metal, strand shell, flux layers, and gap in series.^{[10],[11],[12]}

Many three-dimensional models have been created to model mold heat transfer.^{[1],[2],[3],[4]} Mahapatra showed that to properly convert mold thermocouple temperatures to heat flux, a full 3-D heat flow model of the mold is needed.^{[2],[3]} Thomas used a three-dimensional, finite-element model to predict the thermal and mechanical behavior of slab molds to predict mold temperature, distortion, and residual stress.^[1]

One convention to characterize mold heat transfer is to find the heat transferred from the molten steel to the mold based on the dwell time of the strand against the mold.^[13] This energy corresponds to the shell thickness at mold exit but is rarely checked since the shell thickness is difficult to measure unless there is a breakout failure. Another similar convention to find the total heat transferred to the mold is to measure the cooling water temperature rise. A one-dimensional model converts thermocouple temperatures to discrete heat flux values and a heat flux profile is created that corresponds to the water heat balance.^[14] A typical heat flux curve, that neglects axial conduction, like the one shown in Figure 2.1,^[15] can then be constructed that shows how mold heat transfer varies from the meniscus to the mold exit. A heat flux curve that takes axial heat conduction (heat conduction in the casting direction) into account would have

the maximum heat flux peak just below the meniscus. A one-dimensional heat conduction model, CON1D,^[16] has been developed by Dr. Brian G. Thomas' research group at the University of Illinois at Urbana-Champaign and is the basis for the modeling done throughout this work.

2.1 Introduction to CON1D in-house, slab-casting software

CON1D is software written in FORTRAN by several researchers and graduate students over the years in the Metal Process Simulation Laboratory at the University of Illinois. This software requires minimal set-up time since it takes mold geometry and casting conditions from text input files. It is used to model aspects of steel slab casting including heat transfer and solidification in continuous casting molds. It calculates mold, shell, and water temperatures, shell thickness, interfacial heat flux, ideal mold taper based on 1-D shrinkage calculations, and the thickness and velocity of liquid and solid flux powder.^[16] Another feature is that it can also calculate a variable water-side heat transfer coefficient based on both cooling water velocity and film temperature, and makes estimations of axial heat conduction. Other details, that are often left off of other heat transfer models, that are included in CON1D are the mold plating and the water-scale build up on the inside of the cooling channels.

This paper investigates heat transfer in slab casting molds but may also be relevant to billet casting. Since a billet casting mold is water cooled without the aid of water slots, it serves as a simplified version of a slotted slab casting mold. This simplified example will be the starting basis of the slab mold model.

A 2-D slice through the thickness of a billet mold will be rectangular in shape. Far below the steel meniscus and after the start-up period, the 2-D slice can be modeled having steady-state and constant boundary conditions as shown in Figure 2.2. The mold temperature distribution, T(x), is calculated by solving the general conduction equation, for the steady state case in one dimension.

$$k_{\rm m} \frac{\partial^2 T}{\partial x^2} = 0 \tag{2.1}.$$

A Dirchlet condition, or constant heat flux boundary condition models the heat passing from the interfacial gap to the mold:

$$k_m \frac{\partial T}{\partial x} = q'' \tag{2.2}.$$

The cooling water that flows over the cold face can be modeled by specifying a Neumann condition, or convection boundary condition, on the water channel surfaces expressed as:

$$q'' = h_w (T - T_{water})$$
(2.3).

Figure 2.3 shows the model's equivalent resistance circuit with the resistances to conduction and the resistance to convection in series. The hot and cold face temperatures comprising the end points of the linear solution can be found from the resistance circuit and are:

$$T_{hot,cu} = T_{water} + q_{hot,cu}^{"} \left(\frac{1}{h_w} + \frac{d_m}{k_m} \right)$$
(2.4)

$$T_{cold} = T_{water} + \frac{q_{hot, cu}}{h_w}$$
(2.5).

Since conduction is conservative in one dimension, $q''_{hot,cu}$ is equal to q''_{cold} . The mold thickness, d_m , is a dimension taken from a uniform array of water slots and is the distance from the water slot root to the hot face, not including mold coatings.

Water channels can be added to the model by replacing the water-side heat transfer coefficient, h_w , with an equivalent heat transfer coefficient, h_{fin} , so that the heat removed from the mold to the water is equivalent to that of a solid rectangular slab whose water-side heat transfer coefficient equals some effective value. This effective heat transfer coefficient is based on the water slot geometry by treating the slotted portion of the mold as external heat transfer fins of rectangular profile with adiabatic tips that are mounted to a rectangular slab. Thomas and Ho derived the following effective heat transfer coefficient from extended surface theory where w_{ch} is the channel width, L_{ch} is the channel spacing, d_{ch} is the channel depth, k_m is the mold conductivity, and h_w is the water-side heat transfer coefficient.^[17]

$$h_{fin} = \frac{h_{w}w_{ch}}{L_{ch}} + \frac{\sqrt{2h_{w}k_{m}(L_{ch} - w_{ch})}}{L_{ch}} \tanh\sqrt{\frac{2h_{w}d_{ch}^{2}}{k_{m}(L_{ch} - w_{ch})}}$$
(2.6)^[12]

The first term of equation 2.6 is the contribution from the water slot base, or primary surface, and the second term is the contribution of the fin to the overall fin heat transfer coefficient. Figure 2.4 depicts this analogy between water slots and fins in terms of equivalent boundary conditions and defines mold dimensions. Equations 2.4 and 2.5 can then be modified to include the water slots:

$$T_{hot} = T_{water} + q_{hot,eu}^{"} \left(\frac{1}{h_{fin}} + \frac{d_m}{k_m} \right)$$
(2.7)

$$T_{cold} = T_{water} + \frac{q_{hot,cu}}{h_{fin}}$$
(2.8).

Note the mold thickness, d_m , is the distance from the hot face to the water channel root; it is the thickness of the imaginary slab without mounted fins. The temperature at any position in the mold can then be found by interpolating between these two values can be written with one equation as follows:

$$T_{x} = q'' \left(\frac{1}{h_{fin}} + \frac{x}{k_{m}}\right) + T_{water}$$
(2.9).

 T_x is the temperature at coordinate x. When x equals d_m or 0, T_x is T_{hot} or T_{cold} respectively. This method of finding through-thickness temperatures can be used to compare finite element mold temperatures with CON1D temperatures. When there is one-dimensional conduction, the isotherm plot has straight parallel isotherms and CON1D temperature predictions are expected to match exactly. Equation 2.9 can also be rearranged in the following form to solve for x as shown in equation 2.10.

$$\mathbf{x} = \left[\left(\frac{\mathbf{T}_{\mathbf{x}} - \mathbf{T}_{\text{water}}}{q''} \right) - \frac{1}{\mathbf{h}_{\text{fin}}} \right] \mathbf{k}_{\text{m}}$$
(2.10)

2.1.1 Model boundary conditions

The model domain does not include mold plating since when we assume one-dimensional conduction, the hot-face heat flux incident the copper mold, $q_{hot,cu}$ is equivalent to the interface heat flux, q_{int} , or the heat flux incident the outer mold plating. For simplicity the heat flux will be referred to as q''.

This limit assumes that h_w is constant for any water slot geometry; the parameters that affect h_w like water velocity are adjusted to keep h_w constant. CON1D ^[16] calculates h_w from Sleicher and Rouse's empirical relationship for turbulent flow through a pipe ^[18]:

$$h_{w} = \frac{k_{water}}{D} Nu_{D}$$
(2.10)

$$Nu_{\rm D} = 5 + 0.015 \, {\rm Re}^{\rm a} {\rm Pr}^{\rm b} \tag{2.11}$$

$$D = \frac{4w_{ch}d_{ch}}{2(w_{ch} + d_{ch})}$$
(2.12)

$$a = 0.88 - \frac{0.24}{4 + Pr}$$
(2.13)

$$b = 0.333 + 0.5 e^{-0.6 Pr}$$
(2.14)

$$\operatorname{Re} = \frac{\rho_{\operatorname{water, f}} v_{\operatorname{water}} D}{\mu_{\operatorname{water f}}}$$
(2.15)

Where v_{water} is the water velocity given in meters per second.

$$\Pr = \frac{\mu_{water} Cp_{water}}{k_{water}}$$
(2.16)

$$k_{water} = 0.59 + 0.001 T_{cold}$$
(2.17)

$$\rho_{\text{water,f}} = 1000.3 - 0.040286 \,\mathrm{T_{film}} - 0.0039779 \,\mathrm{T_{film}^2} \tag{2.18}$$

$$Cp_{water} = 4215 - 1.5594 T_{cold} + 0.015234 T_{cold}^{2}$$
(2.19)

$$\mu_{\text{water,f}} = 2.062 \text{ x } 10^{-9} \rho_{\text{water,f}} * 10^{(792.42/_{\text{Tfilm}+273.15})}$$
(2.20)

$$\mu_{\text{water}} = 2.062 \text{ x } 10^{-9} \rho_{\text{water,f}} * 10^{(792.42/_{\text{Tcold}+273.15})}$$
(2.21)

$$T_{\rm film} = \frac{1}{2} (T_{\rm cold} + T_{\rm water})$$
(2.22)

Another commonly used calculation for h_w ^[19] employs the Dittus-Boelter equation ^[20] for the case of cooling in long circular ducts:

$$Nu_{\rm D} = 0.023 \, \mathrm{Re}_{\rm D}^{0.8} \mathrm{Pr}^{0.3} \tag{2.23}.$$

The water properties are taken at the film temperature using equations 2.18, 2.20 for ρ_{water} and μ_{water} and substituting T_{cold} for T_{cold} in equations 2.17 and 2.19 for Cp_{water} and $k_{water.}$.

2.1.2 Adiabatic fin tip assumption

The overall heat transfer coefficient, h_{fin} , was originally formulated for thin fins and assumed adiabatic fin tips. This adiabatic assumption is customary for thin fins since the surface area of the fin tip is much less than that of the rest of the fin. Therefore, the heat escaping from the fin tip is small compared to the heat lost by the fin as a whole. Thick fins do not follow this argument. However, previous analysis by Thomas^[12] showed that thermal distortion causes a gap of 0.1 mm to 0.2 mm to form between the mold, or fin tip, and the steel jacket. This gap can be filled with hard-water scale, air, or water. The approximate upper bound on the heat lost to the fin tip would then be based on a water-filled gap and the lower bound would be based on an air-filled gap. Using a typical fin tip temperature of 40 °C and a steel water jacket temperature of 25 °C, according to Fourier's law and approximating the thermal conductivity of water in the gap as 0.615 Wm⁻¹K⁻¹ and the conductivity of air as 0.027 Wm⁻¹K⁻¹, the range in heat flux through the fin tip is approximately:

$$q'' = k \frac{\Delta T}{\Delta x} = .004 \frac{MW}{m^2} \rightarrow .092 \frac{MW}{m^2}$$

This is small compared with a typical hot-face heat flux range of about 1 to 2.5 MW m⁻².

2.2 Finite element model description

Two-dimensional, steady-state, finite element models, are created to explore the accuracy of the 1-D heat conduction model for various slot widths and spacing and are discussed in section 2.3. The models were created using Ansys 5.5 and map meshed with standard two-dimensional linear isoparametric four-node elements with side lengths of about 0.5 mm. The Ansys designation for these elements is PLANE55. Symmetry was utilized, resulting in an "L" geometry that

represented an idealized, infinite array of fins. A heat flux boundary condition was imposed on the hot face, convection was specified on the fin side surfaces with insulated tips. Each model took about a second to be solved on an SGI Origin2000 computer.

3-D finite element models of typical industrial casting molds were created to further compare the 1-D model temperature predictions and find way to make the 1-D model more accurate. These models were free meshed with linear, thermal, solid, eight-node, brick elements. Each model contained about 10,000 elements and took about 20 seconds to be solved on an SGI Origin2000 workstation. Due to the complex geometry around the thermocouple, solid models of some of the copper molds were created using Pro/Engineer2000, converted to a neutral IGES format and imported into Ansys5.5 such as the Mold #1 wideface and the Mold #2 wide and narrow faces discussed in chapter 3. A constantan stud thermocouple was then added within Ansys for the Mold #1 wideface. The Mold #1 narrowface was created using only the Ansys program.

2.3 Comparison of CON1D and finite element results

The CON1D prediction of mold temperatures using equation 2.9 was checked against an idealized mold with various fin geometries. Five fin geometries were modeled in 2-D and mold temperature results were compared with the 1-D prediction. The temperature profiles for different slot spacing (L_{ch}) and fin thicknesses (L_{ch} - w_{ch}) are presented in Figures 2.5 to 2.10. Two temperature profiles were extracted from each 2-D model, one from a section through the mold thickness that passes through the middle of the fin (section f-f), and another section that passes through the middle of the water slot (section w-w). The fins range from 3 to 19 mm in thickness with various slot widths and spacing. The model parameters for each model are given with each figure. Fins #1 to Fin #5 also had d_m equal to 13 mm, d_{ch} of 12 mm, and k_m of 315 W m⁻¹K⁻¹.

Figures 2.5 and 2.6 present fins #1 and #2 with thickness to length ratios ($(L_{ch}-w_{ch})/d_{ch}$) of 0.25 and 0.33 that show close agreement between the 1-D and 2-D model predictions. The temperature profiles converge linearly and the hot face temperature has a difference of only 1 or 2 °C between the 1-D and 2-D results. The fin heat transfer coefficient, h_{fin}, was calculated for both the 1-D case, using equation 2.6, and the 2-D case. The 2-D h_{fin} was calculated by using the hot face temperature from section f-f and solving for h_{fin} from equation 2.7. This is the same as vertically shifting the 1-D temperature profile so that the hot face temperatures matches. The difference of 1 to 2 °C corresponded to a difference of 3 to 4% between 1-D and 2-D h_{fin} values.

Figure 2.7 shows fin #3 with thickness to length ratio of 0.9, the 1-D and 2-D temperature predictions are still close, resulting in an difference in hot face temperature of only 3 °C and a corresponding difference in predicted h_{fin} values of 5%. Figure 2.8 shows fin #4 with an even greater thickness to length ratio, the temperature predictions have more of a difference corresponding to a greater difference between the 1-D and 2-D h_{fin} values. Finally, Figure 2.9 shows the results for fin #5 with a relatively large thickness to length ratio of 1.5, whose fin heat transfer coefficient is actually lower than the imposed water heat transfer coefficient, shows the most disagreement between the 1-D and 2-D models. The 1-D hot face temperature prediction is 16 °C lower than the 2-D model hot face temperature. The fin length was then increased for the last fin geometry, holding all other dimensions the same as fin #5, shown in Figure 2.9. Both the h_{fin} value and the agreement between the 1-D and 2-D increased. All of the h_{fin} predictions and 2-D model values were compiled into one graph, Figure 2.10 showing the fin thickness ratios as the x-axis. The solid and dashed lines are h_{fin} predictions using equation 2.6. The solid line holds the slot spacing constant and the dashed line holds the slot width constant. The fin thickness ratio

is not the only parameter that defines the slot geometry, therefore, the ratio of slot width to spacing is given as well. As the slot width to spacing ratio (w_{ch}/L_{ch}) approaches unity, the 1-D model is more accurate and the h_{fin} values are higher.

Although the 1-D assumption holds for thin fins, it breaks down for thick fins, where the ratio of slot width to slot spacing, w_{ch}/L_{ch} , is small and where the ratio of slot width to slot depth, w_{ch}/d_{ch} is large . As seen in Figure 2.9, CON1D calculates temperatures about 15 °C or 7%, lower than ANSYS for through thickness temperatures for "thick fins". This difference in temperature can also be thought of as "an offset distance" of about 2 mm. The offset can be used to extract accurate temperature predictions from CON1D by adjusting the depth in the mold where the temperature is examined. An offset of 2 mm means that CON1D results should be examined 2 mm towards the hot face from the point in the mold where a temperature value is desired, in order to find the correct temperature. Finally, the section through the fin has consistently higher ANSYS temperatures than the section through the water slot. The temperatures in both cross sections approach the same value close to the hot face.

2.4 Summary

An existing one-dimensional heat conduction model for mold heat transfer was presented and is based on summing thermal resistances in series. It uses an effective heat transfer coefficient, h_{fin} , based on both water cooling slot geometry and a water-side heat transfer coefficient. The 1-D model is most accurate when the thermal behavior is linear and when the water slot geometry is such that the ratio of slot width and slot spacing approaches unity and the "fin" thickness is small compared to the slot depth.



Figure 2.1 Typical heat flux curve for a slab mold based on mold thermocouple temperatures and a water heat balance



Figure 2.2: Cross section through billet mold and its model with equivalent boundary conditions



Figure 2.3: Equivalent resistance circuit for heat transfer in a billet mold with no water channels.



Figure 2.4: Analogy between water channels and fins allows simplification to equivalent boundary conditions on a rectangular slab.



Figure 2.5: Comparison of 1-D prediction of through-thickness temperatures 2-D model results for an infinite mold with idealized fins (Fin #1)



Figure 2.6: Comparison of prediction of through-thickness temperatures 2-D model results for an infinite mold with idealized fins (Fin #2)

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Figure 2.7: Comparison of prediction of through-thickness temperatures 2-D model results for an infinite mold with idealized fins (Fin #3)

Figure 2.8: Comparison of prediction of through-thickness temperatures 2-D model results for an infinite mold with idealized fins (Fin #4)

Figure 2.9: Comparison of prediction of through-thickness temperatures 2-D model results for an infinite mold with idealized fins (Fin #5)

Figure 2.10: Comparison of prediction of through-thickness temperatures 2-D model results for an infinite mold with idealized fins (Fin #6)

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Figure 2.11: Comparison of 1-D predicted fin heat transfer coefficient, hfin, with 2-D finite element results for Fins #1 - #5 showing the effect of two non-dimensional numbers: wch/Lch and (Lch-wch)/dch on 1-D prediction accuracy and on the value of hfin.

3. ESTIMATING HEAT FLUX FROM MOLD THERMOCOUPLE TEMPERATURE

Temperature readings are often taken with thermocouples that are placed along the axis of a bolt or around other objects. This is unfortunate because this interrupts the usual straight water-slot design and the resulting one-dimensional thermal behavior. Therefore, the determination of heat flux from thermocouple temperature cannot be done accurately assuming one-dimensional conduction alone, as is the current practice in industry.^[5] Figure 3.1 is a three-dimensional isotherm plot for a typical welded stud thermocouple. The thermocouple stud is threaded through the axis of a bolt (omitted from the model) and welded to the copper mold. The temperature is greater above the insulated bolt hole relative to the same depth elsewhere. Thus, the actual heat flux would be lower than that predicted with a one-dimensional result using the inflated temperature reading. This error in heat flux prediction is especially dangerous when the mold geometry changes from one thermocouple to the next in the same mold. However, the onedimensional model can still be used by adjusting the position of the thermocouple in the model so that the model predicts the actual thermocouple temperatures and corresponding actual heat flux. In order to make more accurate heat flux predictions from mold thermocouple temperatures, R.J. O'Malley suggested the use of a correction factor for the one-dimensional model that would account for local three-dimensional thermal behavior around thermocouple sites. This chapter derives the adjustment, called an offset distance, investigates the influence of mold geometry and boundary conditions on the offset distance, and discusses other practical matters pertaining to its implementation. Finally, offsets and temperature to heat flux conversions will be found for typical molds.

3.1 Interpreting thermocouple temperatures using a 1-D model with an offset

As a basis for the offset derivation, expressions for temperature and heat flux that apply to onedimensional conduction will be found. These equations will then be modified to suit the multidimensional case that applies to thermocouple temperatures and their corresponding heat flux boundary condition.

Using mold temperatures to find the hot-face heat flux is simple when the thermal behavior of the mold is truly one-dimensional. The location through the mold thickness where the temperature is known is called the through thickness coordinate, x, as shown in Figure 3.1. The temperature at location x is referred to as T_x . The corresponding heat flux, calculated using this mold temperature, is found by rearranging equation 2.9.

$$q_{hot,cu}'' = \frac{(T_{x} - T_{water})}{\frac{x}{k_{m}} + \frac{1}{h_{fin}}}$$
(3.1)

As discussed in chapter two, the thermal behavior around thermocouples is usually multidimensional. To account for this multidimensionality, the thermocouple position, x, is adjusted in equation 3.1 by an offset distance, d_{offset} . The heat flux expression that can be used with thermocouple temperatures is then given by equation 3.2 where x is replaced by the thermocouple depth, d_{thepl} and increased by d_{offset} .

$$q_{hot,cu}'' = \frac{\left(T_{thcpl} - T_{water}\right)}{\left(\frac{\left(d_{thcpl} + d_{offset}\right)}{k_{m}} + \frac{1}{h_{fin}}\right)}$$
(3.2)

Equation 3.2 collapses to the one-dimensional case, given by equation 3.1, when d_{offset} equals zero. Equation 3.2 can also be written to solve for the thermocouple temperature, T_{thcpl} , which

will be used in section 3.1.3 to compare the accuracy of the equation to 3-D finite element results with various boundary conditions.

$$T_{\text{thepl}} = q'' \left(\frac{1}{h_{\text{fin}}} + \frac{\left(d_{\text{thepl}} + d_{\text{offset}}\right)}{k_{\text{m}}} \right) + T_{\text{water}}$$
(3.3).

The value of the offset distance can be found from a three-dimensional thermal finite element model, choosing reasonable values for the water-side heat transfer coefficient with a water temperature and the hot-face heat flux boundary conditions.

$$\mathbf{d}_{\text{offset}} = \left[\left(\frac{\mathbf{T}_{\text{thcpl},3D} - \mathbf{T}_{\text{water}}}{q''} \right) - \frac{1}{\mathbf{h}_{\text{fin}}} \right] \mathbf{k}_{\text{m}} - \mathbf{d}_{\text{thcpl}}$$
(3.4)

All of the values used in equation 3.4 except the thermocouple temperature, $T_{thcpl,3-D}$, are inputs to the finite element model; $T_{thcpl,3-D}$ is a model output. The heat transfer coefficient, h_{fin} , is calculated using equation 2.6 with the mold geometry, h_w , and k_m from the model. Section 3.1.2 describes the details of the finite element model including how to model the thermocouple and extract the Ansys thermocouple temperature. The first term in equation 3.4 comes from equation 2.10 and is the depth from the cold face where CON1D predicts the same temperature value as the ANSYS thermocouple temperature. The second term is the thermocouple depth from the cold face. An offset of zero means that no adjustment needs to be made since the one-dimensional conduction assumption is valid.

To further illustrate the offset distance, a plot of through thickness temperatures calculated by CON1D and nodal temperatures from a 3-D Ansys model is given in Figure 3.2. The isotherm plot for this 3-D model was previously given in Figure 3.1. Nodal temperatures were extracted from the finite element model in two sections, through a water slot from the cold face to the hot
face (section w-w) and a parallel section that passes through the thermocouple weld point (section t-t). Through thickness temperatures for section w-w were also calculated with the CON1D equation 2.9 for comparison. As seen in Figure 3.1, the left side of the model, around the straight water slots, exhibits one-dimensional thermal behavior as indicated by the straight parallel isotherms. Therefore, the CON1D through thickness temperatures are expected to match the finite element nodal temperatures reasonably well for section w-w. The results plotted in Figure 3.2 show that the temperatures do in fact match.

However, the nodal temperatures through section t-t are much higher than CON1D predicts for the 1-D case due to multidimensional conduction. In order to match the thermocouple temperature an offset distance must be calculated using equation 3.4 and equals 4.5 mm. This is the distance the thermocouple position is shifted toward the hot face in order to match the Ansys thermocouple temperature as shown in the figure. Using this offset distance with equation 3.3, the Ansys thermocouple temperature is of course matched exactly. Figure 3.2 also shows that the difference between the temperature profiles is not constant through the thickness of the mold. Therefore, the offset distance is specified at the thermocouple depth and is only intended to predict thermocouple temperatures.

The offset distance can then be used to find the hot-face heat flux value that corresponds to the plant thermocouple temperature. A heat flux curve, a plot of heat flux vs. distance down the mold in the casting direction, like that shown in Figure $2.1^{[15]}$, can be made from the discrete thermocouple temperatures and corresponding heat flux predictions. The total heat extracted

from the mold can be checked against the cooling water temperature rise that is calculated from the transient CON1D calculation. Details of this water heat balance can be found elsewhere.^[16]

3.1.1 Modeling thermocouples with finite element analysis

Three-dimensional ANSYS finite element models where created to characterize thermal behavior around thermocouple sites in several different mold geometries. Mold geometries are typically unique to each casting plant employing unique slot geometries, spacing, and mold thicknesses. Two molds based on typical mold designs such as the AK Steel (formerly Armco) mold, Mold #1, and a typical geometry for a convectional thick slab casting mold, Mold #2. Plant thermocouple temperatures for Mold #1 and water temperature rise values were found previously^[21] and will be used to validate heat flux predictions. Figure 3.3 is a dimensioned schematic of the Mold #1 wideface model^[21] which employs symmetry boundary conditions on all sides except the hot face, adjacent to the molten steel, and the cold face, subject to water cooling. The top surface, or hot face, is subject to a constant heat flux boundary condition, q". Although the heat flux decreases down the mold length, having its highest value around the meniscus and its lowest value at the mold exit, the relationship between heat flux and thermocouple temperature is easiest to extract by assuming a small section (half of the distance between bolts) in the casting direction has a constant hot-face heat flux. The mold is water cooled. Therefore, a constant convection boundary condition is specified along the inside surfaces of the water slots with a heat transfer coefficient h_w and a constant water temperature, T_{water} . Constant water-side heat transfer coefficients between 30 and 60 kW m⁻¹ K⁻¹, were chosen to represent a range realistically expected in practice. For the same reason, constant water

temperatures were specified between 25 and 35 °C. Bolt holes were left insulated since they are often filled with plastic plugs or air.

Thermocouple placement and type are also unique to each caster. A welded thermocouple with a two millimeter diameter actually conducts a significant amount of heat away from the The same model as shown in Figure 3.1, but excluding the thermocouple weld site. thermocouple, gives a temperature about 6 °C greater at the thermocouple weld site than with the thermocouple. Figure 3.4, a plot of heat flux versus thermocouple temperature, shows that this difference could lead to a 4 % error, or about 0.1 MW m⁻² increase in the predicted hot-face heat flux for that case. The welded thermocouple is modeled to have perfect contact with the mold, so any resistance due to the bonding material is neglected. The material used for the thermocouple is constantan with a constant thermal conductivity of 216 W m⁻¹ K⁻¹. This thermocouple is modeled so it passes through a bolt that passes through the steel water jacket. The bolt is exposed to the cooling water. Therefore, the upper portion of the thermocouple is modeled with an insulated boundary condition while the lower portion has the same convection boundary condition as used in the water slots. Since the thermocouple is not directly exposed to water the result will give a lower bound for the thermocouple temperature. The thermocouple boundary condition profile and the and resulting temperature profile are plotted in Figure 3.5. In practice, the thermocouple is very long so the model includes a long enough portion of the thermocouple for its temperature to become constant. As shown in the figure, the thermocouple temperature approaches the water temperature at its tip, as expected.

3.2 Dependence of offset on boundary conditions

Defining the offset as a distance instead of a temperature rise or by other means is convenient because it was hypothesized that this parameter depended on mold geometry alone. Then only one finite element model need be performed per mold to find the offset unique to its geometry. In this section, the dependence of the offset on boundary conditions is investigated to check our hypothesis. The finite element model of the Mold #1 wideface was run several times, changing the hot-face heat flux, the water-side heat transfer coefficient, and the water temperature. For each case, the offset distance was recalculated. Figure 3.6 shows that there is no change in the offset distance with the hot-face heat flux. Figure 3.7 shows that d_{offset} is also independent of water temperature. However, the offset does depend slightly on the water-side heat transfer coefficient. The previous two figures show offset distances of 4.9 and 4.5 mm for water-side heat transfer coefficients, 35 and 75 kW m⁻² K⁻¹ respectively. The heat flux constant, this range in temperature dependent heat transfer coefficient using the Sleicher-Rouse equation corresponds to about a 12 m/s change in water velocity.

It is important to consider uncertainty in the value of the water side heat transfer coefficient which can be estimated from water velocity and film temperature in a variety of ways. The general form of a heat transfer coefficient is given by equation 2.10 and depends directly on the Nusselt number, Nu_D. The Sleicher-Rouse form for Nu_D was given by equations 2.11 through 2.22 and depends on both the water velocity and film temperature. A simpler form given by the Dittus-Boelter's relationship, equation 2.23 through 2.24 for the Nusselt number typically results in lower approximations of h_w for the same film temperature and water velocity. In practice, however, the film temperature is not a known quantity. What is known is the water temperature,

the mold dimensions and properties, and the thermocouple temperature as given in table 3.1(a) for the Mold #1 wideface. The resulting heat flux predictions, using both equations for h_w, results in lower estimations of the heat transfer coefficient when using the Dittus-Boelter equation and in turn, lower predictions of the hot-face heat flux as shown in table 3.1(b). The difference between the hot-face heat flux predictions using the two methods is about six percent.

The effect of changing h_w on offset distance is explicitly shown in Figure 3.8. There is a slightly negative linear relationship between h_w and offset. As h_w increases, d_{offset} decreases. However, the heat transfer coefficients calculated in typical mold analyses range from about 40 to 60 kWm⁻²K⁻¹. The corresponding range in offset distance is only 0.3 mm. This error in offset corresponds to a small error, about two percent, in the predicted heat flux and varies linearly with heat flux. It is possible to program the linear relationship into the CON1D equations. But, since the error in the predicted heat flux is so small, it is also reasonable to assume that both the waterside heat transfer coefficient and the heat flux are independent of offset so the offset depends only on geometry.

3.3 Dependence of offset on mold thickness

Although mold flux powder aims to reduce friction between the mold surface and strand, surface wear still occurs and increases surface roughness, leading to more wear. Molds are therefore regularly milled about two millimeters, once every few weeks or after a few hundred heats. When a mold has been milled down to minimum thickness, it is then taken out of service. Since the mold thickness changes from week to week, it is therefore important to note its effect on offset distance. Figure 3.9 plots mold through-thickness temperatures for the same mold that has been milled from zero to four times. A new mold is 35 mm thick and is milled two millimeters at a time. The temperature results were taken from three dimensional finite element models based on the Mold #1 mold wideface with identical boundary conditions. The thermocouple temperatures remained almost exactly the same, as did the offset distance. That is not to say that a thick mold will necessarily transfer the same amount of heat as a thin mold. But, the relationship between thermocouple temperature and hot-face heat flux, for a given water-side heat transfer coefficient, remains the same.

3.4 Results: offsets, temperatures, and heat flux curves for commercial molds

This section reports results for two mold sets, Mold #1 and Mold #2 wideface and narrowface. Each of these molds has unique features that shed more understanding on the conversion of thermocouple temperature to the molds hot-face heat flux. For instance, the Mold #1 narrowface does not have a traditional array of uniform water slots. Therefore, the one dimensional temperature equation 2.9 does not match mold temperatures well even far away from the thermocouple. In this case, the offset distance, d_{offset} , can be thought of as containing two parts: one to correct for differences in the hot face temperature prediction, and the other to match the thermocouple temperature. The methodology to find d_{offset} is the same. However, breaking the offset into two parts help explain the high value of d_{offset} in this case.

Mold #2 is unique in the placement of its thermocouple. Both the wideface and the narrowface thermocouples extend beyond the bolt in a thermocouple hole. Since the hole extends beyond the bolt into a more linearly behaving region, the offset distances are quite small.

3.4.1 Mold #1

Mold #1 is based on the AK Steel mold^[21], with separate analysis for the wideface and narrowface. The wideface model is used as an illustration throughout this chapter. The finite element model is given in Figure 3.1 and the dimensioned schematic is given in Figure 3.2. The offset distance is 4.5 mm for the wideface as shown by Figure 3.2. A plot showing the conversion from thermocouple temperature to heat flux using the recommended Sleicher-Rouse equation for figuring the heat transfer coefficient is illustrated in Figure 3.4.

The Mold #1 narrowface model schematic appears in Figure 3.10. Like the wideface model, this model includes a constantan welded stud thermocouple. The boundary conditions are the same as those for the wideface model, employing a constant heat flux boundary condition of 2.5 MW m⁻² on the hot face, constant heat transfer coefficient with a constant water temperature of 54 kW m⁻² K^{-1} and 35 °C along the inner surfaces of the water slots and on the portion of the thermocouple outside the bolt hole. All other surfaces were subject to a zero heat flux condition for reasons previously discussed. The three-dimensional thermal finite element results appear in Figure 3.11 in the form of an isotherm plot. Unlike the wideface, the narrowface does not display one-dimensional thermal behavior around the water slots. This may be explained by the wide span between water slots due to the bolt hole as well as the absence of water slots near the outer edge of the mold. The corner edge of the mold adjacent the angled water slot is the hottest portion of

the mold. Another hot spot occurs adjacent the bolt hole. Some common ways of reducing the temperature of these two hot regions is by deepening the water slots on either side of the bolt hole and the angled water slot. Chapter four describes in detail a method of redesigning an end water slot to attain a desired corner temperature.

The offset for the mold narrowface is 10.3 mm as shown in Figure 3.12 that plots the threedimensional model through-thickness temperatures along with the one-dimensional temperature predictions. This large offset distance can be attributed to the high increase in mold temperature around the thermocouple. In this case, the one-dimensional equations for predicting mold temperatures are also lower than the three dimensional model predicts near the water slots. The offset value can therefore be thought of as containing two parts, a correction for the throughthickness temperatures near the water slots and a correction for multidimensional thermal behavior around the thermocouple. Using this offset value, the relationship between heat flux and thermocouple temperature was calculated implementing the Sleicher-Rouse equation to find the temperature dependent water-side heat transfer coefficient with a constant water velocity of 13.64 m/s. The curve appears as Figure 3.13. Although the points that form the curve were found after a few iterations, the relationship between heat flux and thermocouple temperature is linear. The three-dimensional finite element results, using constant boundary conditions, is plotted for reference and lies close to the curve. The point would have been directly on the curve, had the value of the constant heat transfer coefficient been calculated using the same Sleicher-Rouse relationship.

Thermocouple temperatures for the wideface and the narrowface of Mold #1 were measured^[21] and appear in Figures 3.14 and 3.15 respectively. The heat flux curves were then generated using offsets for both faces and calculating discrete heat flux values. Then the curves were smoothed by balancing the average heat flux with the heat required to increase the cooling water temperature 6.12 °C for the wideface and 7.98 °C for the narrowface. The resulting heat flux curves were given in Figure 2.1^[15] for Mold #1.

3.4.2 Mold #2

The preceding analysis was repeated for Mold #2. The wideface of this mold is unique in that the slot spacing varies from the outer edges to the middle of the mold. The middle wideface slot geometry was chosen for the model. Like the Mold #1 mold analysis, a portion of the mold was modeled with symmetry boundary conditions to represent the repeated identical adjacent sections. The middle portion of the mold contains a slot spacing, L_{ch} , of 15 mm. It is assumed that the middle portion of the mold is far enough away from the outer that the thermal behavior is not affected by the slight change in slot spacing in other parts of the mold. The schematic for the center model appears in Figure 3.16. The boundary conditions are the usual, with a constant heat flux condition on the hot face and constant convection conditions on the inner surfaces of the water slots.

A unique feature of Mold #2 is the small thermocouple hole that extends beyond the bolt hole. Unlike the Mold #1 mold, the thermocouples are pushed into the thermocouple hole that contains a conductive paste. These features were considered in the model and a hand calculation was performed showing that since the conductive paste had a fraction of the conductivity of the mold and thermocouple, the heat lost through the thermocouple was negligible. The decrease in thermocouple temperature in the mold #1 wideface by including a welded thermocouple was 6 $^{\circ}$ C for a typical hot-face heat flux boundary condition of 2.5 MW m⁻² and 54 kW m⁻² K⁻¹ heat transfer coefficient and 35 $^{\circ}$ C water temperature. Since the thermocouple is not welded in this case, the error in thermocouple temperature can be expected to be much lower, even negligible, for this mold.

The isotherm plot for the center wideface model appears in Figure 3.17. The model exhibits onedimensional behavior near the water slots, as indicated by the parallel isotherms in the figure. As mentioned in the Mold #1 analysis, Mold #2 employs a deep water slot on either side of the bolt hole for additional cooling of this hot area. The hottest portion of this mold is still, however, adjacent to the bolt hole. The offset for the center wideface model is 4.2 mm. Plots comparing through-thickness temperatures from the three-dimensional model and the CON1D equations appears in Figures 3.18. As expected, the one-dimensional through-thickness equation matches the three-dimensional model near the uniform array of water slots. Using the calculated offsets, the thermocouple temperatures match the three dimensional model.

Finally, the relationship between heat flux and thermocouple temperature is given by Figure 3.19. The results were generated using the offset found in the previous analysis and by calculating a temperature dependent water-side heat transfer coefficient with constant water velocity of 7.7 m/s. There is a linear relationship between heat flux and thermocouple temperature. However, it should be noted that this relationship is expected to be different for thermocouples placed in regions of the mold where the water slot geometry changes.

It is especially important to regard thermocouple temperatures differently between the narrowface and the wideface for this mold since the mold geometry is so different but also because the water velocity is higher for the Mold #2 narrowface. The water velocity for the mold wideface is 7.7 m/s, whereas the water velocity for the narrowface is higher, 9.8 m/s. Higher water velocities correspond to higher water-side heat transfer coefficients and in turn lower mold temperatures, all else equal.

The narrowface mold model schematic is given in Figure 3.20. Like the Mold #1 mold, the narrowface employs an angled water slot near the edge of the mold. It also has the same thermocouple placement scheme with the small diameter thermocouple hole that protrudes beyond the bolt hole. The boundary conditions are the same as in the wideface model. The finite element temperature results are given in Figure 3.21. The mold corner has the highest temperature. The isotherms also bend around the bolt hole and the thermocouple hole indicating multidimensional thermal behavior. The offset distance for this mold is 4.0 mm as shown by Figure 3.22, a plot comparing the through thickness temperatures from the three dimensional model to those predicted by CON1D. Finally, Figure 3.23 shows the relationship between heat flux and thermocouple temperature for a constant water velocity of 9.8 m/s. A thermocouple temperature of 105 °C corresponds to 1.8 MW m⁻² heat flux. Therefore, the difference in predicted heat flux for the same thermocouple temperature for the center wideface and the narrowface is about 10 percent.

3.5 Summary

A correction to the one-dimensional heat conduction equations given in chapter two was developed. This correction, called an offset distance, is virtually independent of thermal boundary conditions and is primarily based on the mold geometry around the thermocouple site. The offset distance is the distance the thermocouple position is shifted in the one-dimensional model so that thermocouple temperatures correspond to the proper hot-face heat flux boundary condition. Without this correction to the 1-D model, heat flux predictions associated with inflated thermocouple temperatures would be too high. Since cooling slot geometry is not always constant in the same mold face, and usually not the same between the wide and narrow faces, is especially important not to compare thermocouples in the same mold without considering the effect of mold geometry on the reading. Three dimensional finite element models were used to find the offset distances associated with four models representing two industrial casters mold narrow and wide faces.

d _{ml}	dm	d_{ch}	L_{ch}	Wch	d _{thcpl}	doffset	k _m	$V_{\rm W}$	T_{water}	T _{thepl}
mm							$W m^{-1} K^{-1}$	m/s	°C	
35	30	5	21.5	16	6	4.5	315	11.67	35	130

 Table 3.1(a): Mold #1 wideface dimensions and other known values

Table 3.1(b): Calculated values including heat flux using two methods for
calculating hw for the Mold #1 wideface.

	h _w	$\mathbf{h}_{\mathrm{fin}}$	Nu _D	T _{hot}	T _{cold}	q _{hot}
W	m	$^{-2} \mathrm{K}^{-1}$		°C		MW m ⁻²
Dittus-Boelter	41797	45263	494.5	236	73	1.71
Sleicher-	48905	52285	564.9	242	70	1.81
Rouse						
% difference	15					6



 $q = 2.5 \text{ MW m}^{-2}$

Figure 3.1: Three-dimensional thermal behavior around bolt hole for a representative mold face (mold #1 wideface)



Figure 3.2: Through-thickness mold temperature profiles comparing 3D Ansys and CON1D predicitons near the water slots and near the thermocouple



Figure 3.3: Dimensioned schematic of Mold#1 wideface model including thermocouple



Figure 3.4: Relationship between heat flux and thermocouple temperature for Mold #1 wideface mold given various ways to calculate the water-side heat transfer coefficient used in the CON1D equations.



Figure 3.5: Temperature profile of a welded constantan thermocouple in the Mold #1 wideface.



Figure 3.6: Dependence of offset on heat flux boundary condition for two water side heat transfer coefficient , h_w, values.



Figure 3.7: Dependence of offset on cooling water temperature for two constant water-side heat transfer coefficients



h_w, Water-side heat transfer coefficient [W m ⁻² K⁻¹]

Figure 3.8: Dependence of offset on water-side heat transfer coefficient for constant water temperature and heat flux for Mold#1 wideface

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Figure 3.9: Effect of mold milling due to wear on offset distance for Mold#1 wideface



all dims in [mm]



Figure 3.10: Model schematic for mold #1 narrowface



Figure 3.11: 3-D thermal finite element results for Mold #1 narrowface



Figure 3.12 Through-thickness mold temperature profiles comparing 3D Ansys and CON1D predicitons near the water slots and near the thermocouple for the Mold#1 narrowface



Figure 3.13: Relationship between heat flux and thermocouple temperature for Mold #1 narrowface calculated using the CON1D equations, one point given from a 3-D model.



Figure 3.14: Thermocouple temperatures for Mold #1 wideface and the corresponding temperature profile associated with the heat flux curve in Figure 1.1. (from literature)^[Z]



Figure 3.15: Thermocouple temperatures for Mold #1 narrowface and the corresponding temperature profile associated with the heat flux curve in Figure 1.1. (from literature)^[Z]



Figure 3.16: Dimensioned schematic of the Mold#2 wideface model



Figure 3.17: 3-D thermal finite element results for Mold #2 wideface



Figure 3.18: Through-thickness mold temperature profiles comparing 3D Ansys and CON1D predicitons near the water slots and near the thermocouple for the Mold#2 wideface



Figure 3.19: Relationship between heat flux and thermocouple temperature for Mold #2 wideface calculated using the CON1D equations, one point given from a 3-D model.



Figure 3.20: Dimensioned schematic of the Mold#2 narrowface model



Figure 3.21: 3-D thermal finite element results for Mold #2 narrowface



Figure 3.22: Through-thickness mold temperature profiles comparing 3D Ansys and CON1D predicitons near the water slots and near the thermocouple for the Mold#2 narrowface



Figure 3.23: Relationship between heat flux and thermocouple temperature for Mold #2 narrowface calculated using the CON1D equations, one point given from a 3-D model.
4. ESTIMATING NARROWFACE CORNER TEMPERATURE

Large temperature gradients along the surface of a continuous casting mold lead to non-uniform thermal expansion, residual stress, distortion, and decreased mold life due to frequent remachining. Physical constraints from gasket seals prevent the extension of the standard straight water slot design to the edge of the mold narrowface. This edge is therefore subject to higher temperatures than the rest of the face. Figures 4.1 and 4.2 show the thermal distortion exaggerated 50-fold that results from typical non-uniform mold cooling from the literature.^[1] The top corner of the narrowface has the highest temperature causing the mold faces to distort in such a way that only a small line of contact between the wide and narrowface results. The high clamping forces between the two faces is thought to lead to permanent crushing of the narrowface corner and can lead to a steel fin defect or even create a sticking corner breakout failure.^[1]

Because creep corresponds to a reduction of strength at high temperatures, it is important to maintain reasonable temperatures in the critical corner region. Many mold designs attempt to reduce the corner temperature by including an angled end water slot. The objective of this chapter is to gain insight into the design of end water slots in order to achieve desired corner temperatures and a uniform temperature mold hot face.

4.1 Model Description

A series of models based on the Mold #1 narrowface were created with varying end slot design in order to investigate the effects of end slot dimensions on the hot face corner temperature. The models were all two-dimensional, steady state, finite element heat transfer models. A representative domain is shown in Figure 4.3. Boundary conditions, shown in Figure 4.4, were as follows:

 \cdot along the internal edges of the water slots, a convection boundary condition , q=h_w(T-T_{water}) was specified where T_{water} is the cooling water temperature, h_w is the water heat transfer coefficient, and T is the resulting mold temperature,

• along the center of the narrowface at x=0, y=0, a symmetry boundary condition, q=0, allowing one half of the narrowface to represent the whole,

 \cdot and finally, along the outside edge where the narrowface contacts the wideface, an insulated boundary condition was used. It is conservative to assume q=0 here since the heat flux would be very low on this face; specifying zero heat flux will yield slightly higher temperatures on the hot edge which we wish to reduce in temperature. The cold face, including the portions interrupted by water slots and the left edge, also has a zero heat flux boundary condition.

In addition to the boundary conditions and the given geometry, the following constant conditions, shown in the table below, were specified:

Thermal conductivity of copper, km	$350 \text{ W m}^{-1} \text{ K}^{-1}$
Water slot heat transfer coefficient, h_W	$35,000 \text{ W m}^{-2} \text{ K}^{-1}$
Water temperature, Twater	30 °C

Two-dimensional finite element models were created using Pro/Engineer2000, converted to a neutral IGES file, and then imported into Ansys5.5. The models were free-meshed with standard four node isoparametric bilinear elements, Ansys designation PLANE57. The elements were of unit thickness (1 mm) but with planar conduction and insulated faces so that they acted as 2-D

elements. The finite element mesh was created with free area mapping usually at level three refinement. This setting corresponds to elements with about 1 mm sides. Each model had about 1200 nodes resulting in computational time of a second or two on an HP C240 workstation. Detailed specifications of the mold dimensions and heat flux values for each model along with tabulated numerical results are presented in Table 4.1.

4.2 Results

Typical results for the two-dimensional, quarter mold, Ansys finite element models in this study are given in Figure 4.5 along with corresponding conditions. The model results are presented as colored isotherms superimposed over the mold geometry. As expected, the coldest isotherms appear near the water slots and the material farthest away from the water slots has the highest temperature. A typical feature is the high temperature (here 216.5 °C) of the hot face corner adjacent to the angled water slot. Additionally, the temperature along the internal water slot surface typically exceeds the water temperature by ~10 to 60 °C for various molds outlined in Table 4.1.

4.2.1 Nondimensional results

Corner temperatures were recorded for a series of models with 25 mm thick molds and eight end slot designs and imposed hot-face heat fluxes ranging from 0.5 to 10 MW m⁻². The corner temperatures are plotted in Figure 4.6 versus L_c , the shortest distance from the angled water slot edge to the corner. There is a roughly linear relationship between L_c and the corner temperature, T_c , for each heat flux value. As L_c increases so does the corner temperature. Increasing the imposed heat flux value increases both the temperature and the slope of each group of tests. The

effect of heat flux on the corner temperature can be perfectly captured by expressing the corner temperature as a non-dimensional number, T^{*}:

$$T^{*} = \frac{k_{m}(T_{c} - T_{w})}{q_{hot,cu}^{"}L_{eff}}$$
(4.1).

Where the effective length, L_{eff} is defined

$$L_{eff} = \frac{k_m}{h_{fin}} + d_m$$
(4.2)

and the fin heat transfer coefficient is given in equation 2.6. The constants in equations 4.1 and 4.2 are given in Table 4.1 and in the two-dimensional model schematic, Figure 4.3. When the results in Figure 4.6 are non-dimensionalized with respect to corner temperature, all of the results collapse to one line as shown in Figure 4.7.

Not only does the heat flux condition and water slot design vary in practice, but so does the mold thickness, d_{ml} . Figure 4.8 illustrates the effect of mold thickness on hot face temperature. The dimensionless corner temperature is plotted against L_c for 25 mm, 35 mm, and 45 mm thick molds. Four end slot geometries, cases 1, 3, 5, and 7, were used with each mold thickness. A thicker mold corresponds to a larger L_c for each water slot geometry. Since L_{eff} is smaller for thinner molds, and is in the denominator for T^* , the non-dimensional temperature is inflated for thickness by non-dimensionalizing the corner distance, L_c , to L^* , given by:

$$L^* = \frac{L_c}{d_m}$$
(4.3)

The results from Figure 4.8 were normalized as a function of mold thickness, and appear in Figure 4.9. Again, the results collapse to a single line.

Figure 4.10 shows dimensionless temperature results from several models as a function of dimensionless length. The data is fit to one linear curve:

$$T^* = 0.655 L^* + 0.420 \tag{4.4}$$

where T^* and L^* denote dimensionless temperature and length respectively. Equation 4.4 can be solved for T_c :

$$T_{c} = \left(0.655 \frac{L_{c}}{d_{m}} + 0.420\right) \left(\frac{q}{h_{fin}} + \frac{d_{m}}{k_{m}}\right) + T_{w}$$
(4.5).

This relationship to predict the corner temperature, based on a corner distance, does not account for the orientation of the slot. Requisite parameters include the slot angle, the distance to the hot face, L_x , and the distance to the insulated edge, L_y . Figure 4.10 shows the corner temperatures of a series of four models with the same L_c but varying angle, A. The greater the angle, the lower the non-dimensional temperature for a 25 mm mold thickness and constant L_c of 12.95 mm as shown in models m25_a30, m25_a45, m25_a50, and m25_a60. The model number designation "m25" stands for a mold of thickness, d_{ml} , equal to 25 mm. The designation "a" stands for end slot angle, followed by the angle in degrees. A close-up of the previous figure shows this effect of slot angle on corner temperature for these four models in Figure 4.11. The difference in varying the slot angle \pm 15° about 45° results in about a \pm 6 % change in T^{*}. Holding L_c and boundary conditions constant, equation 4.5 over-predicts T_c for a slot with a lower L_y since the slot is closer to the hot face. This difference in T^{*} corresponds to about \pm 8 °C for a typical 2 MW m⁻² heat flux in a 25 mm thick mold with L_c equal to 13 mm. Greater accuracy likely warrants 2-D consideration.

4.3 Implementation

To find an end slot design that results in a uniform hot face temperature the following steps should be implemented. First, the desired uniform hot face temperature is specified. If the water slot geometry not including the end slot is a uniform array of slots with equal spacing between slots, then the hot face temperature can be accurately calculated using equation 2.7. If the mold geometry is not uniform, and the thermal behavior is not expected to be one-dimensional, a more accurate way of finding the hot face temperature is through a finite element model. The optimal L_c can be found based on the specified hot face temperature which is used as the corner temperature, T_c . Rearranging equation 4.5 yields

$$L_{c} = \left((T_{c} - T_{w}) \frac{k_{m}}{qL_{eff}} - 0.420 \right) \frac{d_{m}}{0.655}$$
(4.6).

This dimension is the only one that the method specifies. Common practice is to use the same slot width as the other slots . This method was used to redesign the Mold #1 narrowface end slot. Figure 4.11 shows an optimized hot face temperature design. This mold has a corner temperature of 189 °C which compares well with the mid slot hot face temperature of 185 °C. An unavoidable dip in temperature occurs between these two points caused by the angled slot. This model also shows a very high temperature hot face across from the bolt hole due to the larger slot spacing there.

4.4 Summary

Hot face temperature variations are known to lead to quality problems like fin defects as well as surface quality problems. These quality problems are a very complex subject that needs further investigation. The results show that the distance between the hot face corner and the closest water slot, L_c , has a significant influence on the corner temperature. The closer the water slot is to the corner, the lower the corner temperature. The corner temperature can be predicted in degrees Celsius based on the slot array and mold geometry by equation 4.5:

$$T_{c} = \left(0.655 \frac{L_{c}}{d_{m}} + 0.420\right) \left(\frac{q}{h_{fin}} + \frac{d_{m}}{k_{m}}\right) + T_{w}$$
(4.5).

This relationship can be used to roughly estimate the corner temperature of casting mold narrowfaces if an angled end slot is used. This simple example provides easy tool to help design narrowface slots to achieve desired hot face temperatures.

Constants		test code	explainati	<u>no</u>							
dch = 0.012 m		mYYZC									
$hw = 35000 \text{ W} \text{ m}^{-2} \text{ K}^{-1}$		YY=mold	thickness,	d _{ml} [mm]							
$km = 315 \text{ W m}^{-1} \text{ K}^{-1}$		Z=test nur	nber referr	ing to the p	aramtric	study of	angled w	ater slot w	idth, angle an	d height.	-
Lch = 0.011 m		C= additic	onal inform	ation	{TXX=	distance f	rom angl	ed water s	lot to corner 1	efered to a	s Lc [mm],
Twater = $30 ^{\circ}$ C wch = 0.005m					aXX= 6	and water	slot angl	e [degrees	~		
	1										
	Tc	Lc	dm	dml	A	W	Η	Leff	þ	T*	hfin
test code, "filename"	[°C]	[m]	[m]	[m]	[₀]	[m]	[m]	[m]	[W m ⁻²]	Ŀ	$[W m^{-2} K^{-1}]$
m25_1_q2	225.0	0.0227	0.013	0.025	80	0.004	0.011	0.0195	2.0E+06	1.5734	48,329.5
m25_2_q10	1,051.0	0.0241	0.013	0.025	80	0.009	0.011	0.0195	1.0E+07	1.6478	48,329.5
m25_3_q2	195.0	0.0190	0.013	0.025	80	0.004	0.016	0.0195	2.0E+06	1.3318	48,329.5
$m25_{-}4_{-}q10$	886.1	0.0204	0.013	0.025	80	0.009	0.016	0.0195	1.0E + 07	1.3817	48,329.5
m25_5_q2	195.5	0.0181	0.013	0.025	45	0.004	0.011	0.0195	2.0E+06	1.3358	48,329.5
m25_6_q10	958.7	0.0215	0.013	0.025	45	0.009	0.011	0.0195	1.0E+07	1.4989	48,329.5
m25_7_q2	154.6	0.0111	0.013	0.025	45		0.016	0.0195	2.0E+06	1.0058	48,329.5
m25_8_q10	706.0	0.0144	0.013	0.025	45		0.016	0.0195	1.0E+07	1.0910	48,329.5
m25_a30	172.1	0.01295	0.013	0.025	30			0.0195	2.0E+06	1.1468	48,329.5
m25_a45	163.0	0.01295	0.013	0.025	45			0.0195	2.0E+06	1.0736	48,329.5
m25_a50	161.1	0.01295	0.013	0.025	50			0.0195	2.0E+06	1.0578	48,329.5
m25_a60	155.4	0.01295	0.013	0.025	60			0.0195	2.0E+06	1.0117	48,329.5
m25_a45_L8	135.1	0.00800	0.013	0.025	45			0.0195	2.0E+06	0.8478	48,329.5
m25_a45_L17	187.3	0.01700	0.013	0.025	45			0.0195	2.0E+06	1.2693	48,329.5
m25_a45_L22	219.6	0.02200	0.013	0.025	45			0.0195	2.0E+06	1.5301	48,329.5
m35_1_q2	271.8	0.02994	0.023	0.035	80	0.004	0.011	0.0295	2.0E+06	1.2901	48,329.5
m35_3_q2	246.4	0.02544	0.023	0.035	80	0.004	0.016	0.0295	2.0E+06	1.1546	48,329.5
m35_5_q2	252.6	0.02661	0.023	0.035	45	0.004	0.011	0.0295	2.0E+06	1.1876	48,329.5
m35_7_q2	224.4	0.02011	0.023	0.035	45	0.004	0.016	0.0295	2.0E+06	1.0374	48,329.5
m35_5_q10	1, 143.0	0.02661	0.023	0.035	45	0.004	0.011	0.0295	1.0E+07	1.1877	48,329.5
m35_L10_a60	158.8	0.01000	0.023	0.035	60	0.004	0.026	0.0295	2.0E+06	0.6874	48,329.5
m35_L30_a60	270.3	0.03000	0.023	0.035	60	0.004	0.009	0.0295	2.0E+06	1.2824	48,329.5

Table 4.1(a): Results, conditions, and geometry of a series of 2-D Ansys finite element models

	Тс	Lc	dm	dml	Α	M	Η	Leff	б	T*	hfin
test code, "filename"	[^o C]	[m]	[m]	[m]	[]	[m]	[m]	[m]	[W m ⁻²]	[-]	[W m ⁻² K ⁻¹]
m45_1_q2	328.7	0.03850	0.033	0.045	80	0.004	0.011	0.0395	2.0E+06	1.1903	48,329.5
m45_3_q2	306.9	0.03368	0.033	0.045	80	0.004	0.016	0.0395	2.0E+06	1.1037	48,329.5
m45_5_q2	313.7	0.03592	0.033	0.045	45	0.004	0.011	0.0395	2.0E+06	1.1306	48,329.5
m45_7_q2	291.7	0.02976	0.033	0.045	45	0.004	0.016	0.0395	2.0E+06	1.0430	48,329.5
m45_L13_a70	184.9	0.01300	0.033	0.045	70	0.004		0.0395	2.0E+06	0.6172	48,329.5
$m45_L20_a70$	233.8	0.02000	0.033	0.045	70	0.009		0.0395	2.0E+06	0.8124	48,329.5

Table 4.1(b): Results, conditions, and geometry of a series of 2-D Ansys finite element models

Table 4.1(c): Results and heat flux for a series of 2-D Ansys finite element models testing the effect of various constant heat flux conditions, q, given in W m⁻² geometry same as in Table 3.2(a)

	A = X D	S OF+OS	$^{-\Lambda}$	$2 0E\pm0.6$	$C = X^{U}$	2 0E±06	$V = X^{D}$	S OF+06	$= \Lambda^{D}$	1 OF+07
5		0.01.00	-vh	7.UL 100	-vh	0. AL 00	, vy	00.100	-vh	1.01.10/
T _c	[T^*	T_{c} [^o C]	T^{*}	T_{c} [^o C]	T^*	T_c [^o C]	T^*	T _c [^o C]	T^*
	8.75	1.5734	224.99	1.5734	322.48	1.5734	517.46	1.5734	1005.00	1.57357
\sim	81.07	1.6483	234.26	1.6483	336.40	1.6483	540.66	1.6483	1051.00	1.6478
Ì	71.58	1.3420	195.04	1.3318	277.56	1.3318	442.61	1.3318	855.21	1.3318
	72.81	1.3817	201.22	1.3817	286.84	1.3817	458.06	1.3817	886.12	1.3817
	71.38	1.3357	195.53	1.3358	278.29	1.3358	443.82	1.3358	857.65	1.3358
`	76.44	1.4989	215.75	1.4989	308.62	1.4989	494.37	1.4989	958.75	1.4989
•	51.16	1.0058	154.64	1.0058	216.95	1.0058	341.59	1.0058	653.18	1.0058
-	53.80	1.0910	165.20	1.0910	232.81	1.0910	368.01	1.0910	706.02	1.0910



Figure 4.1: Thermally distorted interface between the narrow and wide faces showing the line of contact which may lead to a crushed corner showing 50 times exagerated distortion (from literature)²



Figure 4.2: Three-dimensional view of narrowface and wideface contact region showing thermal distortion exaggerated 50 fold (from literature)²



Figure 4.3: Representative two-dimensional finite element heat conduction model schematic shown with variable and constant dimensions.



Figure 4.4: Two-dimensional mold Ansys finite element model schematic with boundary conditions. Internal water slot edges have convection condition, other edges on cold face are insulated.



Figure 4.5: Typical isotherms for the series of models performed in this study.



Figure 4.6: 2-D finite element narrow face corner temperature as a function of Lc, measured for eight angled water slot designs using 25 mm thick molds with various constant hot face heat flux boundary conditions.



Figure 4.7: Non-dimensional corner temperature that was normalized with hot face heat flux boundary condition plotted as a function of Lc



Figure 4.8: Dimensionless corner temperatures for four end slot designs with varying mold thicknesses, dml, plotted versus Lc.



Figure 4.9: Dimensionless corner temperature of four end slot designs with varying mold thicknesses as a function of normalized Lc.



Figure 4.10: Normalized 2-D finite element corner temperatures plotted for various thickness molds and end slot designs as a function of a dimensionless length.



Figure 4.11: Uniform hot face temperature design for the model #1 narrowface [°C]

5. CONCLUSIONS

Molds are routinely instrumented with thermocouples that give only part of the information needed to characterize mold temperatures and heat flux. It has been previously shown that a 3-D model is needed to capture the relationship between heat flux and the resulting thermocouple temperatures. However, 3-D computations require long preparation times and are not convenient for most casting plants to perform. Therefore, a simple model is needed to characterize mold heat transfer and mold temperatures from thermocouple data.

The 1-D heat conduction equation that predicts mold through-thickness and face temperatures,

$$T_{x} = q'' \left(\frac{1}{h_{fin}} + \frac{x}{k_{m}}\right) + T_{water}$$
(2.9)

has been validated for both idealized slab molds, with the usual repeated straight water slot design, and for four commercial molds. It is most accurate for molds with water-slot width to spacing ratios (w_{ch}/L_{ch}) close to unity and for small "fin thickness" to slot depth ratios ($L_{ch}-w_{ch}$)/d_{ch}.

This one-dimensional model has been enhanced in order to make more accurate interpretations of thermocouple temperatures to predict heat flux using a correction factor called an offset distance

$$\mathbf{d}_{\text{offset}} = \left[\left(\frac{\mathbf{T}_{\text{thcpl},3D} - \mathbf{T}_{\text{water}}}{\mathbf{q}''} \right) - \frac{1}{\mathbf{h}_{\text{fin}}} \right] \mathbf{k}_{\text{m}} - \mathbf{d}_{\text{thcpl}}$$
(3.4).

The offset distance accounts for multidimensional thermal behavior around thermocouple sites. and is the equal to the distance the thermocouple position in the mold is theoretically shifted toward the mold hot face so that the actual heat flux corresponds to mold thermocouple temperatures in the 1-D model. This offset distance is dependent primarily on mold geometry and is virtually independent of all boundary conditions including the heat flux, water-side heat transfer coefficient, and the cooling water temperature. The offset distance, d_{offset}, is calculated using the results from one steady-state, three-dimensional, thermal, finite element model of the desired mold geometry. Since d_{offset} is calculated using the imposed boundary conditions from the model, the mold and cooling water dimensions, mold thermal conductivity, and the resulting thermocouple temperature, it is simply a distillation of the 3-D results. Discrete heat flux values can then be accurately estimated from their corresponding mold thermocouple temperatures using the following relation:

$$q_{hot,cu}'' = \frac{\left(T_{thcpl} - T_{w}\right)}{\left(\frac{\left(d_{thcpl} + d_{offset}\right)}{k_{m}} + \frac{1}{h_{fin}}\right)}$$
(3.2).

A heat flux curve, like that shown in Figure 2.1,^[15] can then be constructed using the discrete heat flux values, assuming of course that the average heat flux matches a heat balance on the cooling water temperature rise. Without this correction to the 1-D model, heat flux predictions associated with inflated thermocouple temperatures would be too high.

Offset distances were found for both the wideface and the narrowfaces for a commercial mold design based on the AK Steel mold (Mold #1) and a typical thick slab casting mold (Mold #2). The offsets for the Mold #1 wideface and narrowface were 4.5 mm and 10.3 mm respectively. Resulting heat flux curves based on these offsets were also shown for Mold #1 based on mold

thermocouple temperatures. The offsets for the Mold #2 wideface and narrowface were 4.2 mm and 4.0 mm respectively.

Since narrowface corner temperatures are associated with several quality problems such as excessive mold distortion, creep, corner crushing, resulting fin defects, and strand surface quality, a simple way to estimate this corner temperature has been developed. Narrowface corner temperatures from several two-dimensional, thermal, finite element models with varying end slot geometry have been distilled into a simple relationship based on mold geometry and boundary conditions:

$$T_{c} = \left(0.655 \frac{L_{c}}{d_{m}} + 0.420\right) \left(\frac{q''}{h_{fin}} + \frac{d_{m}}{k_{m}}\right) + T_{w}$$
(4.5).

This relationship uses mold thermal boundary conditions, including heat flux and an effective heat transfer coefficient, h_{fin} , and is based on the mold cooling slot geometry including an angled end slot. The results show that the distance between the hot face corner and the closest water slot, L_c , has a significant influence on the corner temperature. The closer the water slot is to the corner, the lower the corner temperature. This simple relationship provides an easy tool to help design narrowface slots to achieve desired hot face temperatures without the time needed for the set up and execution of a 2-D computation.

Both of these simple relationships, to estimate the narrowface corner temperature and to relate thermocouple temperatures to heat flux, can be incorporated into the CON1D in-house software. This enhanced 1-D conduction model will give more accurate mold temperature results than the steady-state, 3-D, finite element model since it is transient, calculates a variable water-side heat

transfer coefficient based on both cooling water velocity and film temperature, and makes good estimations of axial heat conduction. The preparation time is also less for the CON1D software when compared to a 3-D finite element model since mold geometry is readily input as a separate text file.

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