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# Modeling heat transfer and skull-clogging in submerged entry nozzles

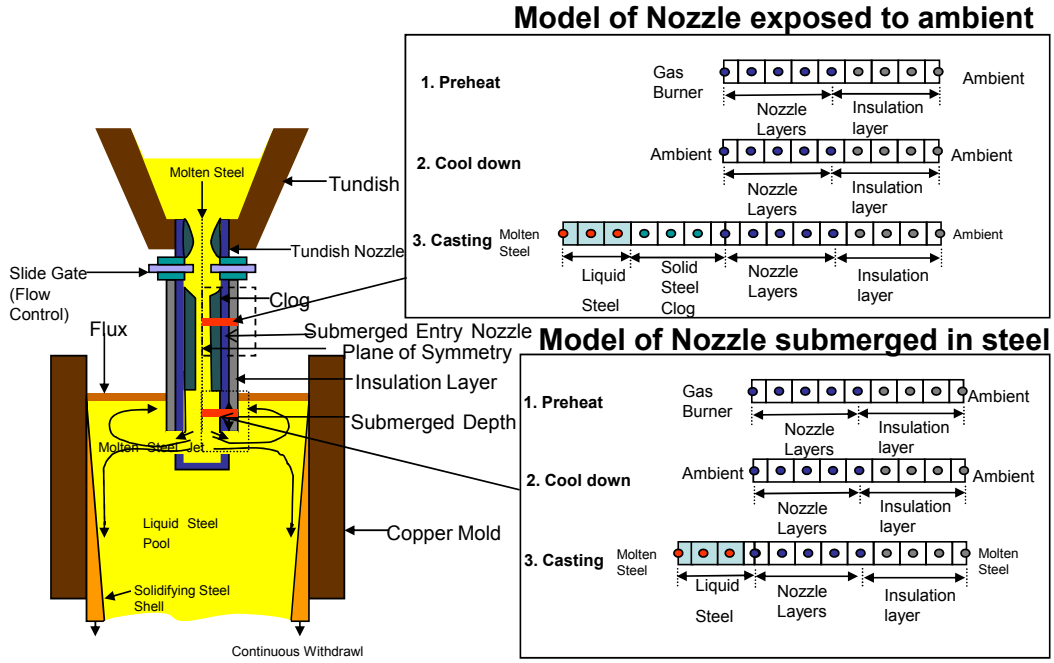
**Varun Kumar Singh, B.G. Thomas**

*Department of Mechanical Science and Engineering*
**University of Illinois at Urbana-Champaign**

## Objectives

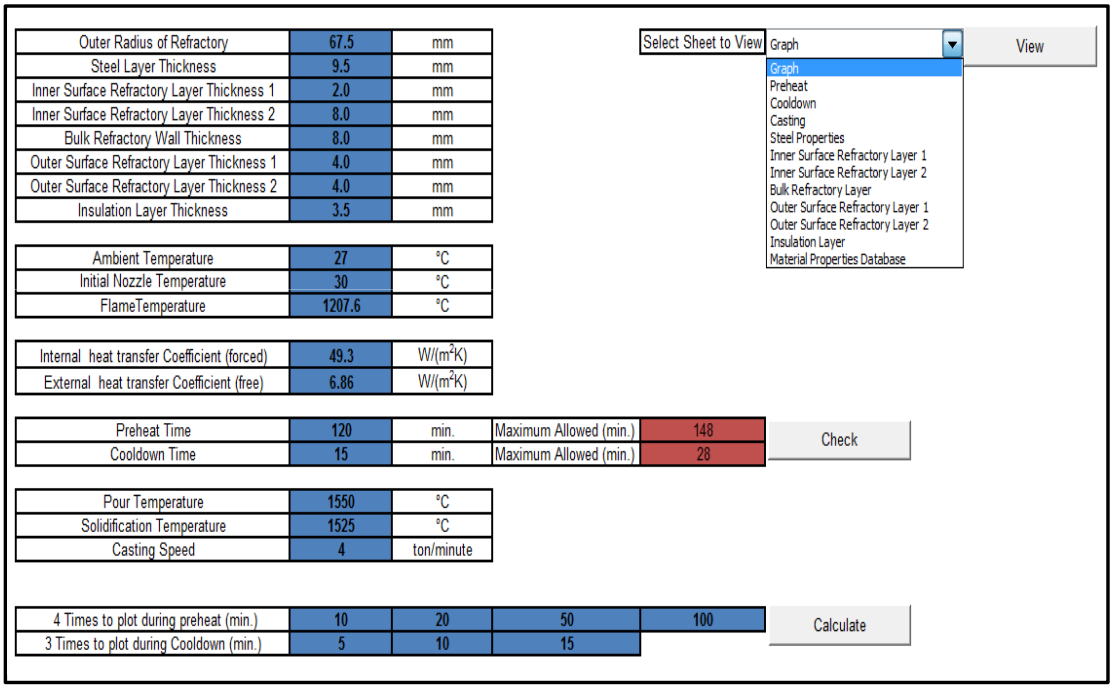
- Develop a spreadsheet based tool to calculate the heat transfer coefficients and flame temperature during preheating of the nozzle.
- Develop a spreadsheet based tool to model the heat transfer in submerged entry nozzles during the three stages: preheat, cool down and casting.
- Compare the heat transfer characteristics of two refractory materials: doloma graphite and alumina graphite

# Model & computational domain






# Main page of the tool



| | | |
|---|---|---|
| Outer Radius of Refractory | 67.5 | mm |
| Steel Layer Thickness | 9.5 | mm |
| Inner Surface Refractory Layer Thickness 1 | 2.0 | mm |
| Inner Surface Refractory Layer Thickness 2 | 8.0 | mm |
| Bulk Refractory Wall Thickness | 8.0 | mm |
| Outer Surface Refractory Layer Thickness 1 | 4.0 | mm |
| Outer Surface Refractory Layer Thickness 2 | 4.0 | mm |
| Insulation Layer Thickness | 3.5 | mm |

| | | |
|---|---|---|
| Ambient Temperature | 27 | °C |
| Initial Nozzle Temperature | 30 | °C |
| FlameTemperature | 1207.6 | °C |

| | | |
|---|---|---|
| Internal heat transfer Coefficient (forced) | 49.3 | W/(m$^2$K) |
| External heat transfer Coefficient (free) | 6.86 | W/(m$^2$K) |

| | | | | |
|---|---|---|---|---|
| Preheat Time | 120 | min. | Maximum Allowed (min.) | 148 |
| Cooldown Time | 15 | min. | Maximum Allowed (min.) | 28 |

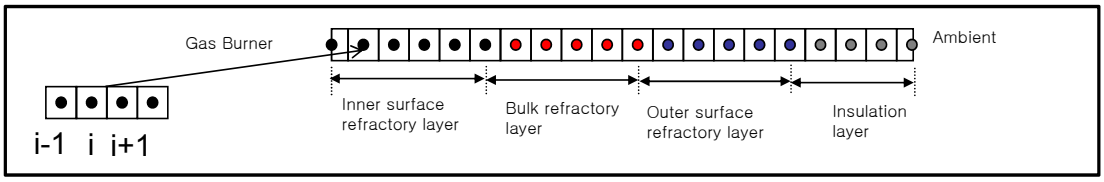
Check

| | | |
|---|---|---|
| Pour Temperature | 1550 | °C |
| Solidification Temperature | 1525 | °C |
| Casting Speed | 4 | ton/minute |

| | | | | |
|---|---|---|---|---|
| 4 Times to plot during preheat (min.) | 10 | 20 | 50 | 100 |
| 3 Times to plot during Cooldown (min.) | 5 | 10 | 15 | |

Calculate

# Governing Equation and finite difference equation for interior nodes




- Heat conduction equation in cylindrical co–ordinates [1]

$$\rho C_p \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)$$

- Using Taylor series [1] expansion, the equation is discretized as:

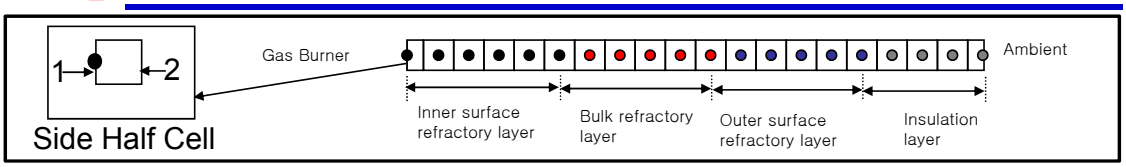
$$\Rightarrow \quad \rho C_p \frac{\partial T}{\partial t} = \frac{k}{r}\left[\frac{\partial T}{\partial r} + r\frac{\partial^2 T}{\partial r^2}\right] \qquad \Rightarrow \quad \frac{\partial T}{\partial t} = \alpha\left[\frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2}\right]$$

$$\Rightarrow \quad \frac{T_i^{n+1} - T_i^n}{\Delta t} = \alpha\left[\frac{1}{r}\frac{T_{i+1}^n - T_{i-1}^n}{2\Delta r} + \frac{T_{i+1}^n - 2T_i^n + T_{i-1}^n}{\Delta r^2}\right]$$

$$\Rightarrow \quad T_i^{n+1} = T_i^n + \alpha\Delta t\left[T_{i+1}^n\left(\frac{1}{\Delta r^2} + \frac{1}{2r\Delta r}\right) + T_{i-1}^n\left(\frac{1}{\Delta r^2} - \frac{1}{2r\Delta r}\right) - \frac{2T_i^n}{\Delta r^2}\right]$$



# Finite Difference Equations (Side nodes with convection)



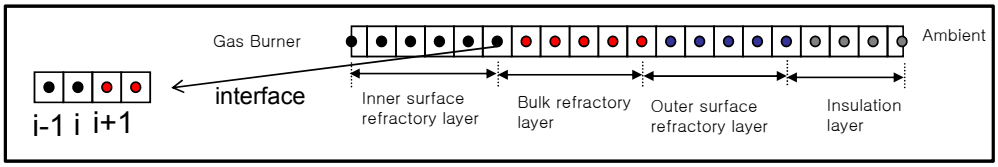

- Heat balance on side half cell gives:

$$\rho C_p V \frac{\partial T}{\partial t} = -kA_2 \frac{\partial T}{\partial r}\Big|_2 + hA_1 \Delta T\Big|_1$$

$$\Rightarrow \quad \rho C_p \frac{r\Delta r}{2}\frac{T_i^{n+1} - T_i^n}{\Delta t} = k\left(r + \frac{\Delta r}{2}\right)\left(\frac{T_i^n - T_{i+1}^n}{\Delta r}\right) + hr\left(T_{ambient} - T_i^n\right)$$

$$\Rightarrow \quad T_i^{n+1} = T_i^n + \frac{2\alpha\Delta t}{r\Delta r}\left(r + \frac{\Delta r}{2}\right)\left(\frac{T_i^n - T_{i+1}^n}{\Delta r}\right) + \frac{2\Delta thr}{\rho C_p r\Delta r}\left(T_{ambient} - T_i^n\right)$$

# Finite Difference Equations (Interface Nodes)




$$\rho C_p \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)$$

$$\Rightarrow \rho C_p \frac{T_i^{n+1} - T_i^n}{\Delta t} = \frac{1}{\Delta r r_i}\left[r_{i+1/2}k_2 \frac{\partial T}{\partial r}\Big|_{i+1/2} - r_{i-1/2}k_1 \frac{\partial T}{\partial r}\Big|_{i-1/2}\right]$$

$$\Rightarrow \rho C_p \frac{T_i^{n+1} - T_i^n}{\Delta t} = \frac{1}{\Delta r r_i}\left[\left(r + \frac{\Delta r}{2}\right)k_2 \frac{T_{i+1}^n - T_i^n}{\Delta r} - \left(r - \frac{\Delta r}{2}\right)k_1 \frac{T_i^n - T_{i-1}^n}{\Delta r}\right]$$

$$\Rightarrow T_i^{n+1} = T_i^n + \frac{\Delta t}{r_i \Delta r^2}\left[\alpha_2\left(r + \frac{\Delta r}{2}\right)(T_{i+1}^n - T_i^n) - \alpha_1\left(r - \frac{\Delta r}{2}\right)(T_i^n - T_{i-1}^n)\right]$$



# Steel Shell Solidification Model

- Enthalpy formulation of the transient 1-D heat conduction equation is solved:

$$\rho \frac{\partial H}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)$$

- Top row temperatures:

$$T_i = T_{pour}$$

- Top row enthalpies [2]:

$$H_i = C_p * T_{pour} + L_f * \mathrm{int}\left(\frac{T_{pour}}{T_{solidus}}\right)$$

where $L_f$ is the latent heat of fusion.

# Steel Shell Solidification Model

- Enthalpy of interior nodes:

$$H_i^{n+1} = H_i^n + \frac{k\Delta t}{\rho}\left[T_{i+1}^n\left(\frac{1}{\Delta r^2}+\frac{1}{2r\Delta r}\right)+T_{i-1}^n\left(\frac{1}{\Delta r^2}-\frac{1}{2r\Delta r}\right)-\frac{2}{\Delta r^2}T_i^n\right]$$

- Enthalpy of side nodes with convection:

$$H_i^{n+1} = H_i^n + \frac{2h\Delta t}{\rho\Delta r}\left[T_{steel}-T_i^n\right]+\frac{2k\Delta t}{\rho r\Delta r}\left(r+\frac{\Delta r}{2}\right)\left(\frac{T_i^n - T_{i+1}^n}{\Delta r}\right)$$

- After the enthalpy has been calculated the temperatures are then calculated using [2] :

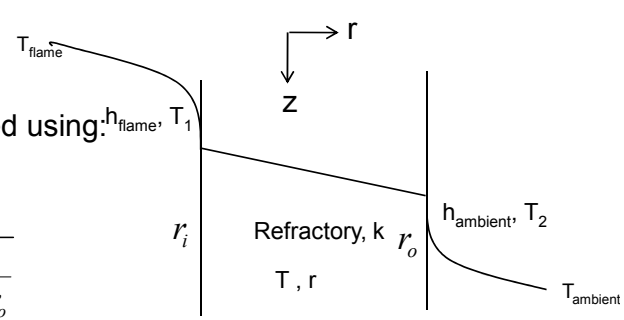
$$T_i = \min\left[\frac{H_i}{C_p}, \max\left\{\frac{H_i - L_f}{C_p}, T_{sol}\right\}\right]$$



# Validation of Steady State Aspect of the Model

- Compared the results of the simulation when it reaches steady state with analytical Solution.
- Governing equation for Analytical solution [1]:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)=0$$




- Heat flux through the nozzle is calculated using:

$$q = \frac{T_{flame} - T_{ambient}}{\frac{1}{h_{flame}r_i}+\frac{\ln(r_o/r_i)}{k}+\frac{1}{h_{ambient}r_o}}$$

$$r_i = r_o - t$$

- Finally, the temperatures in the nozzle are :

$$T_1 = T_{flame} - \frac{q}{h_{flame}r_i}$$

$$T_2 = T_{ambient} + \frac{q}{h_{ambient}r_o}$$

$$T = T_1 - \frac{q}{k}\ln\left(\frac{r}{r_i}\right)$$

# Simulation conditions for validation of steady state aspect of the model

| Label | Symbol | Value | Units |
|---|---|---|---|
| Outer Radius of Refractory | $r_o$ | 67.5 | mm |
| Bulk Refractory Wall Thickness | t | 29.5 | mm |
| Initial Nozzle Temperature | $T_{intital}$ | 27* | °C |
| Ambient Temperature | $T_{ambient}$ | 27 | °C |
| Flame Temperature | $T_{flame}$ | 1460 | °C |
| Internal Convection heat transfer Coefficient (Forced) | $h_{flame}$ | 50 | $W/(m^2K)$ |
| External Convection heat transfer Coefficient (Free) | $h_{ambient}$ | 7.3 | $W/(m^2K)$ |
| Thermal Conductivity | K | 18.21 | W/m-K |
| Specific Heat | $C_p$ | 804* | J/kg-K |
| Density | ρ | 2347 | $kg/m^3$ |
| Stefan Boltzman's Constant | σ | 5.67E-8 | |
| Emmissivity | ε | 0.96 | |

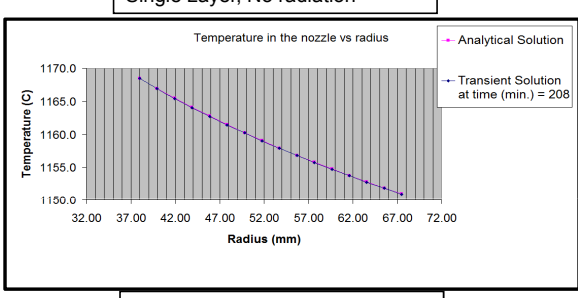
* Parameters required by transient simulation method



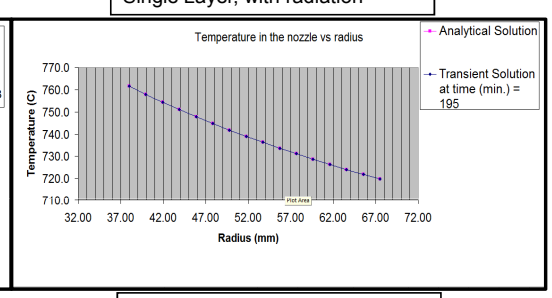
# Validation – Steady state aspect of the model

- The results of the simulation are in good agreement with that of the analytical solution

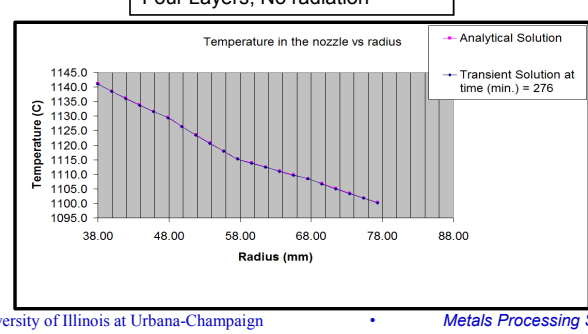
Single Layer, No radiation



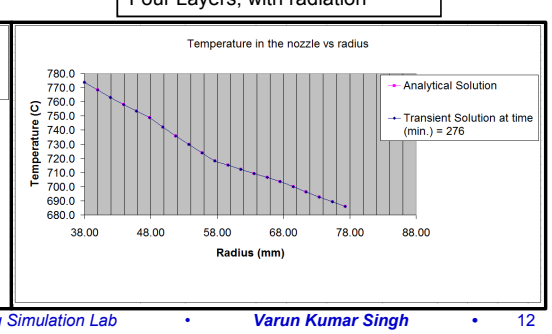

Single Layer, with radiation




Four Layers, No radiation




Four Layers, with radiation



# Validation of transient aspect of the model

- Compare the results of the simulation with that of the lumped thermal heat capacity model.
- System undergoing a transient thermal response to a heat transfer process has a nearly uniform temperature and small differences of temperature within the system can be ignored.
- The model is valid only if the Biot number (hL/k) < 0.1
- The governing equation is [1]

$$\rho V C_p \frac{dT}{dt} = -hA(T - T_e)$$

- To solve this equation, one initial condition is required:
  t=0: $T=T_o$.
  Solving the equation, the temperature at any time,t can be calculated from:

$$\frac{T - T_e}{T_o - T_e} = e^{-(hA/\rho V C_p)t}$$

where $T_o$ is the initial surface temperature, $T_e$ is the ambient temperature.

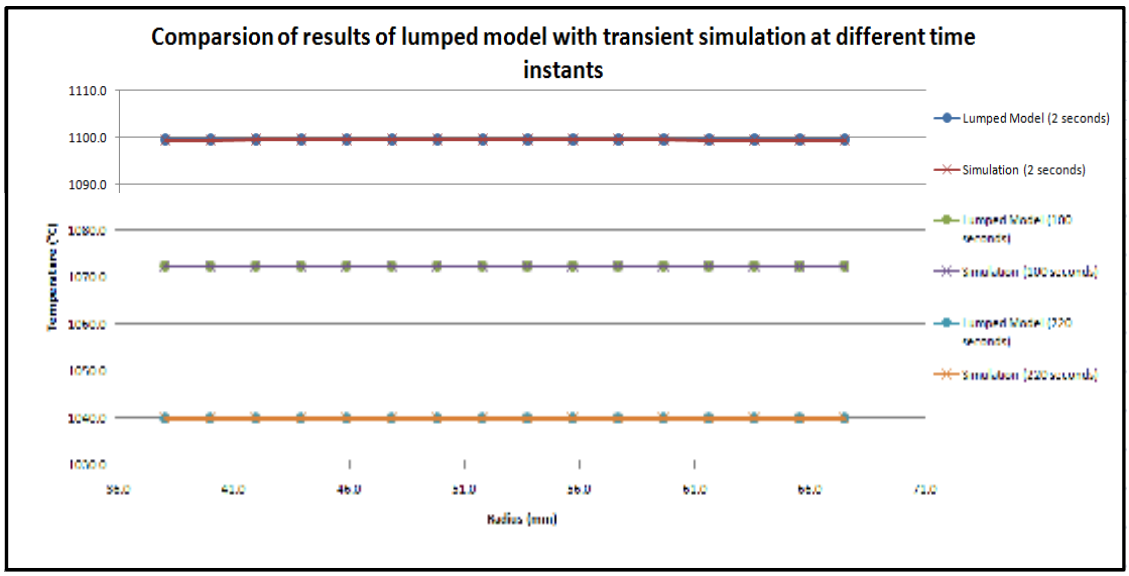


# Simulation Parameters – Validation of transient aspect of the model

| Label | Symbol | Value | Units |
|---|---|---|---|
| Outer Radius of Refractory | $r_o$ | 67.5 | mm |
| Bulk Refractory Wall Thickness | t | 29.5 | mm |
| Initial Nozzle Temperature | $T_{intital}$ | 1100 | °C |
| Ambient Temperature | $T_{ambient}$ | 27 | °C |
| External Convection heat transfer Coefficient (Free) | $h_{ambient}$ | 7.3 | W/(m²K) |
| Thermal Conductivity | K | 1000 | W/m-K |
| Specific Heat | $C_p$ | 804 | J/kg-K |
| Density | ρ | 2347 | kg/m³ |
| Stefan Boltzman's Constant | σ | 5.67E-8 | |
| Emmissivity | ε | 0.96 | |

# Comparison of Results of lumped model and transient simulation

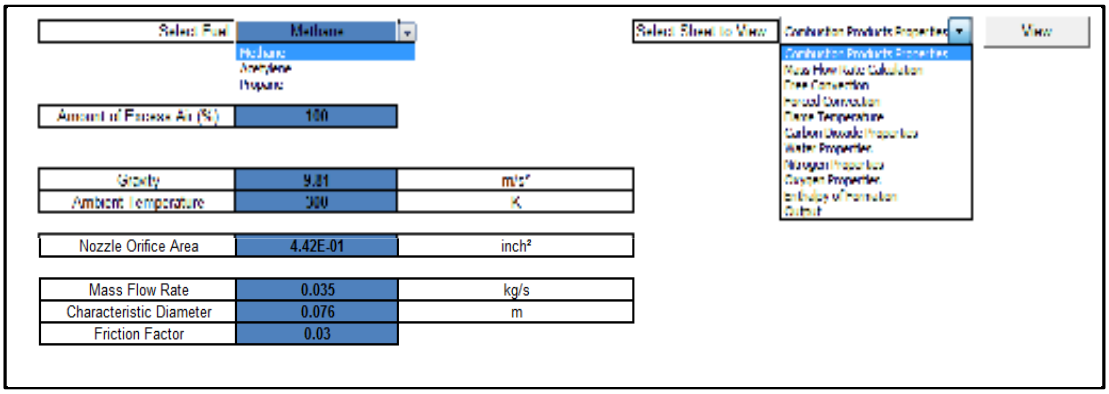
• The results of the simulation are in good agreement with that of the lumped thermal heat capacity model






# Calculation of Flame temperature and heat transfer coefficients

Input Page of the tool




• It is assumed that the fuel is burning in air.
• The user is asked to select the fuel and amount of excess air relative to the stoichiometric amount.
• The nozzle orifice area is required to calculate the mass flow rate.
• The mass flow rate sheet of the tool calculates the mass flow rate for range of pressures. The user should select the mass flow rate desired and input on this page.

# Properties of mixture of gases in combustion products

• Thermal conductivity of the mixture of gases is calculated using Saxena and Mason [3]:

$$\lambda_m = \sum_{i=1}^{n} \frac{y_i \lambda_i}{\sum_{j=1}^{n} y_j A_{ij}}$$

Where $\lambda_m$ = the thermal conductivity of the gas mixture
$\lambda_i$ = the thermal conductivity of pure i
$y_i, y_j$ = mole fractions of component i and j

$$A_{ij} = \frac{\left[1 + (\eta_i / \eta_j)^{1/2} (M_j / M_i)^{1/4}\right]^2}{\left[8 (1 + M_i / M_j)\right]^{1/2}}$$

$$A_{ji} = \frac{\eta_j}{\eta_i} \frac{M_i}{M_j} A_{ij}$$

Where $\eta_i, \eta_j$ are the viscosities of pure i and j respectively

And $M_i, M_j$ are the molecular weights of pure i and j

• Thermal diffusivity, kinematic viscosity, density and specific heat are calculated using the particle mixture rule.



# Combustion reaction and flame temperature calculation

- Stoichiometric Reaction

$$CH_4 + 2O_2 + 7.5N_2 \rightarrow CO_2 + 2H_2O + 7.5N_2$$

- Reaction with 100 % excess air

$$CH_4 + 4O_2 + 15N_2 \rightarrow CO_2 + 2H_2O + 15N_2 + 2O_2$$

- Flame temperature calculation [4]

The enthalpy of formation of products is balanced with that of the reactants:

$$\sum_P n_e \overline{h_e} = \sum_R n_i \overline{h_i} \Rightarrow \sum_P n_e (\overline{h_f^\circ} + \Delta\overline{h})_e = \sum_R n_i (\overline{h_f^\circ} + \Delta\overline{h})_i \Rightarrow \sum_P n_e (\Delta\overline{h})_e = \sum_R n_i (\Delta\overline{h})_i + \sum_R n_i \overline{h_{fi}^\circ} - \sum_P n_e \overline{h_{fe}^\circ}$$

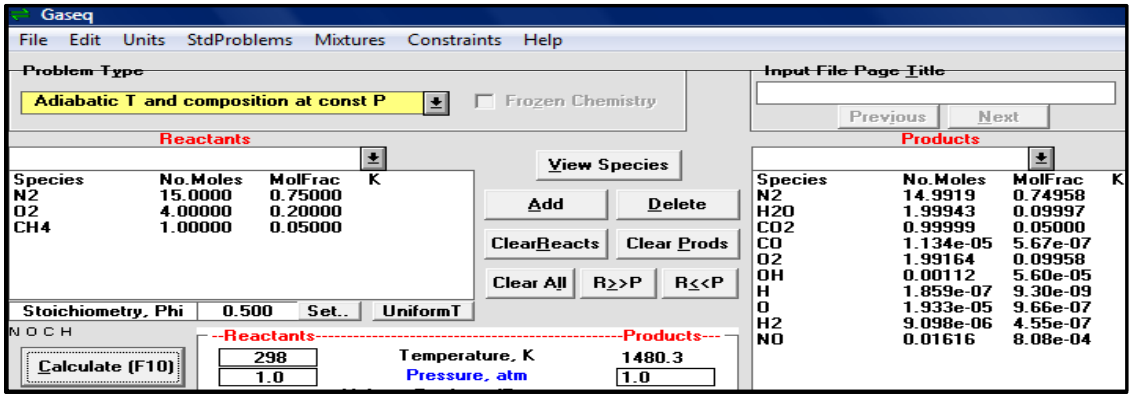
$$\Rightarrow \sum_P n_e (\Delta\overline{h})_e = \sum_R n_i \overline{h_{fi}^\circ} - \sum_P n_e \overline{h_{fe}^\circ}$$

$$\Rightarrow (\overline{\Delta h})_{CO_2} + 2(\overline{\Delta h})_{H_2O} + 7.5(\overline{\Delta h})_{N_2} = \left[(\overline{h_f^\circ})_{CH_4} + 2\cancelto{0}{(\overline{h_f^\circ})_{O_2}} + 7.5\cancelto{0}{(\overline{h_f^\circ})_{N_2}}\right] - \left[(\overline{h_f^\circ})_{CO_2} + 2(\overline{h_f^\circ})_{H_2O} + 7.5\cancelto{0}{(\overline{h_f^\circ})_{N_2}}\right]$$

- Balancing the enthalpy of formation in the above equation gives the flame temperature

# Validation of flame temperature calculation

- The tool was run for methane fuel and 100 % excess air.
- The results were compared with a commercial software Gaseq.
- The flame temperature calculated from the tool is 1480.6 K
- The results from the tool are shown in the figure below:




- The two tools are found to be in good agreement.



# Heat transfer coefficient calculation

- Free Convection to ambient:
  - The Churchill and Chu [1] equation for flow over a vertical flat plate is used

$$Nu_{avg} = \left\{ 0.825 + \frac{\left[0.387\, Ra^{1/6}\right]}{\left[1 + \left(\frac{0.429}{Pr}\right)^{9/16}\right]^{8/27}} \right\}^2$$

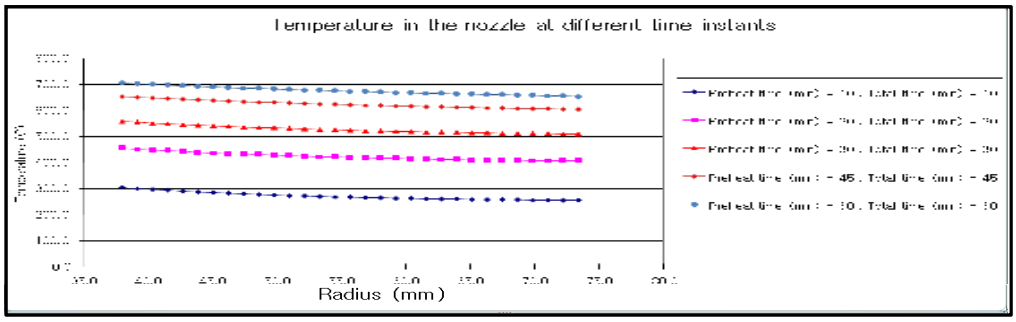
- Forced Convection from flame:

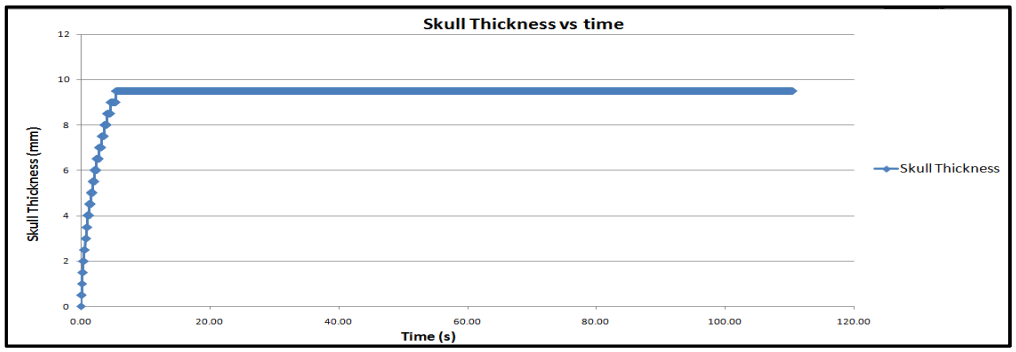
  The Petukhov, Kirillov, and Popov [1] is used

$$Nu = \frac{\left[(f/8)\,\mathrm{Re}_D\,\mathrm{Pr}\right]}{\left[1.07 + 12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)\right]}$$

- The forced heat transfer coefficient was calculated to be 49.3 W/m$^2$K.
- The free heat transfer coefficient was calculated to be 6.9 W/m$^2$K

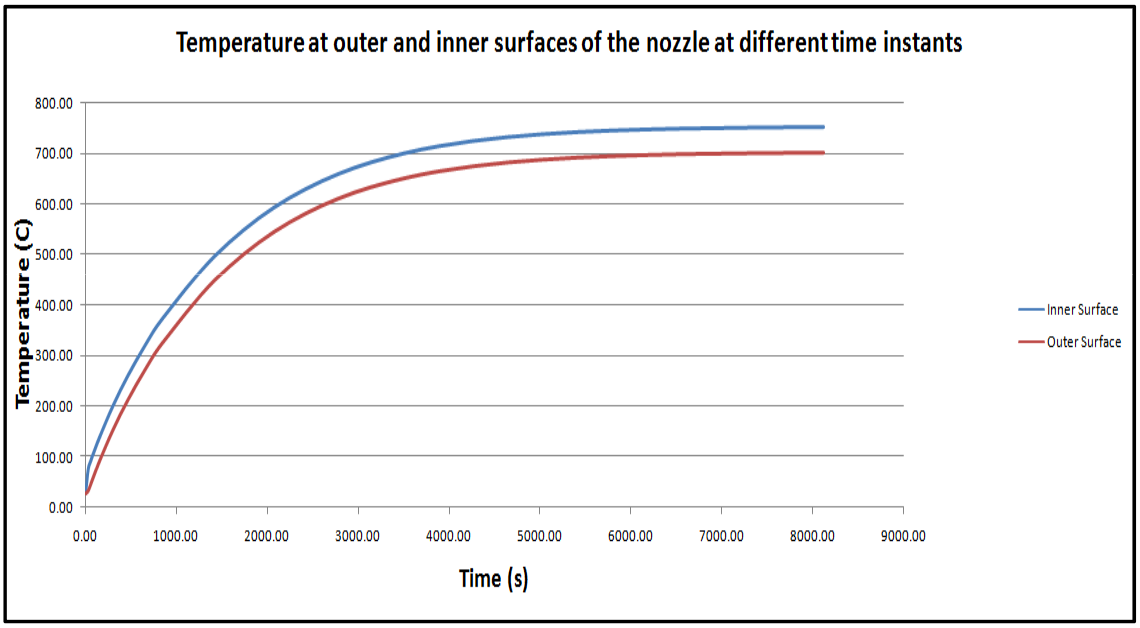
# Output page of the tool






# Temperature variation with time



# Conclusions

- Alumina graphite nozzle reaches 5% higher temperature than doloma graphite (700 vs 670 °C) after equal preheat time of 1hour), so skulling is about the same.
- Air entrainment should be decreased, because excess air reduces the flame temperature.
- The mass flow rate of the reactants should be increased to have a higher heat transfer coefficient, thereby increasing nozzle temperatures after preheat.
- Steady state is not reached even after an hour of preheating of a nozzle (with 30-mm thick wall). The preheating time should be increased.



# References

1. Incropera, F.P., and Dewitt, P.D., 2002, *Fundamentals of Heat and Mass Transfer,* John Wiley and Sons, New York.
2. Thomas, B.G. and B. Ho, "Spread Sheet Model of Continuous Casting," Journal of Engineering for Industry, ASME, New York, NY, Vol. 118, No. 1, 1996, pp. 37-44.
3. Poling, B.E., O'Connell J.M., and Prausnitz, J.M., 2001, *The Properties of Gases and Liquids,* McGraw-Hill, New York.
4. Moran, J.M., and Shapiro, H.N., 2004, *Fundamentals of Engineering Thermodynamics,* John Wiley and Sons, New York.
5. Report on "Modeling of Tundish Nozzle Preheating" by J. Mareno and B.G. Thomas.

# Acknowledgements


- Continuous Casting Consortium Members (ABB, Arcelor-Mittal, Baosteel, Corus, Delavan/Goodrich, LWB Refractories, Nucor, Nippon Steel, Postech, Steel Dynamics, ANSYS-Fluent)
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- Rajneesh Chaudhary and other Graduate students at Metals Processing Simulation Laboratory, UIUC.