High Emissivity Coatings – Do they Really Work?*

J.W. Stendera, S. Bonsall

There is little doubt that the subject of emissivity and what effect it has on the thermal efficiency of practical industrial furnaces is a matter of some confusion and contention. This paper sets out to discuss the theoretical effect of furnace lining emissivity on the heating rate of a load and through experimentation with a lab scale steel reheat furnace to confirm the calculations and demonstrate the practical result.

1 Introduction

First results of an experiment rich in information on furnace lining emissivity on the heating rate of a load are presented based on comprehensive calculations and practical studies in a lab furnace. The applied radiant heat transfer theory is explained and its application on furnace heating is monitored.

2 Experimental method

A gas-fired furnace was constructed to simulate a steel reheat furnace as shown schematically in Fig. 1 and in a photograph in Fig. 2. A thermocouple was installed into a drilled hole at the center of five-inch (127 mm) cubes of carbon steel. The furnace was pre-heated to 1093 °C (2000 °F) and the block was quickly inserted into the furnace. The heat up rate of the block and other parameters were monitored with a data logger.

*Tested by AZ Technology, Huntsville, AL, by their method to determine total hemispherical emittance at selected temperatures based on ASTM E408

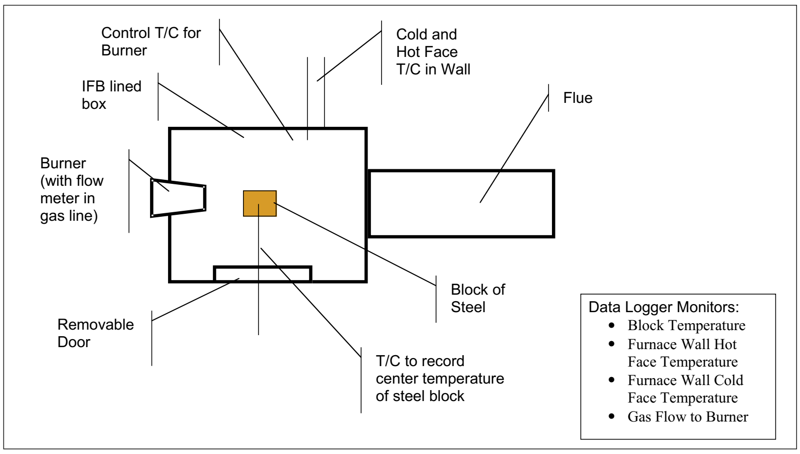

Fig. 1 Scheme of experimental set-up

The experiment was run first with uncoated insulating firebrick walls of emissivity 0,49* at 1093 °C (2000 °F) and then repeated after being coated with a SiC mortar with an emissivity of 0,89*. The heat-up rate of the block was a measure of potential gains in productivity (faster throughput) and the gas flow was a measure of any improvements in fuel efficiency. Wall thermocouples were to determine the effect of the coating on heat transfer through the furnace wall. There are many effects that need discussion and this topic likely deserves many papers. For brevity, only the theoretical calculations based on the literature are discussed and the results are compared to the actual heat up of the steel block.

Fig. 2 Experimental set-up

3 Radiant heat transfer theory

NASA sparked interest in high emissivity coatings in the 1960's and 1970's for use in spacecraft. The association with NASA gave the whole subject an aura of "high tech" and mystery. NASA's interest surrounded the need to improve the removal of heat from space vehicles while in orbit and during re-entry. In the vacuum of outer space, the only heat transfer mechanisms available are radiation and mass transfer. Mass transfer was used on the original craft that orbited the earth and went to the moon in the form

Jim W. Stendera
Sam Bonsall
Vesuvius R&D Center
44815 Bettsville, OH
USA

Corresponding author: *Jim Stendera*
E-mail: jim.stendera@us.vesuvius.com

Keywords: thermal efficiency, industrial furnaces, emissivity

*Paper presented at ACerS St. Louis meeting 2011. Acknowledgement to our cooperation partner ACerS for passing the publication rights.

of the ablative coating on the re-entry shield. With a reusable spacecraft this method was not practical which left emissivity as the only choice. Space itself is extremely cold, as low as 3 K (–270 °C). Even in our solar system, where there are more energetic particles present than in the coldest parts of space, the temperature is only about 40 K (–233 °C). Therefore when a space vehicle is even at a very moderate temperature there is still sufficient temperature difference for radiant heat transfer to taken place according to the Stefan-Boltzmann equation:

$$q = A\varepsilon\sigma(T_h^{\,4} - T_c^{\,4}) \qquad (1)$$

where,
q: net power radiated from the object [W]
ε: emissivity of the object
σ: Stefan-Boltzmann constant = $5{,}6703 \cdot 10^{-8}$ W/m²K⁴
T_h: hot body absolute temperature [K]
T_c: cold surroundings absolute temperature [K]
A: surface area of the object [m²]

For example, two objects in space at 300 K (27 °C/81 °F) and 1 m² surface area will radiate the following power depending on their emissivity:
object 1: ε = 0,9; 413 W
object 2: ε = 0,5; 230 W
The higher emissivity object will radiate 1,8 times the power of the lower emissivity object. When both objects are at a higher temperature such as 800 K (527 °C), the power radiated is as follows:
object 1: ε = 0,9; 20 902 W
object 2: ε = 0,5; 11 613 W
Once again, the higher emissivity object radiates 1,8 times the power of the lower emissivity object. This is simply because all the values substituted into the Stefan-Boltzmann equation are the same except for emissivity. The ratio of the emissivities is 0,9 / 0,5 or 1,8. The total power radiated for both objects increases by more than 50 times at the higher temperature however, showing the effect of the fourth power relation to temperature while maintaining the 1,8 ratio relationship.

4 Radiant transfer theory applied to furnaces

The published papers on high emissivity coatings use the Stefan-Boltzmann equation to show that the coatings on a furnace wall will increase heat energy transferred to a cold load (such as a steel slab), but do not attempt to estimate on a theoretical basis how much more energy will be transferred. Even a casual comparison to the radiation in space scenario leads to the conclusion that the case of heating a colder object in a furnace is much different. First, there are now two emissivity values to consider, that of the hot furnace walls and that of the cold object. Second, unless the furnace is a vacuum furnace radiation is not the only mechanism of heat transfer.

4.1 Heat transfer between two objects of different emissivity

The case of two objects with differing emissivities is considered first. A review of a heat transfer textbook [1] shows that this case can get complicated very quickly. In general, the net radiative heat transfer (Q_{net}) between two surfaces, 1 and 2, at temperatures T_1 and T_2 depends on the following:
- T_1 and T_2
- areas of both surfaces, A_1 and A_2
- the shape, orientation and spacing of 1 and 2
- the radiative properties of the surfaces
- additional surfaces in the environment, whose radiation can be reflected from one surface to the other
- the medium between the surfaces. If it absorbs, emits or "reflects" radiation (when the medium is air these effects can usually be neglected).

If the surfaces are black, if they are surrounded by air (or vacuum), and no heat flows between them by conduction or convection, then only the first three considerations are involved in determining Q_{net}. This leads to an expression of the Stefan-Boltzmann equation as follows:

$$Q_{net} = A_1 F_{1-2}\sigma(T_1^{\,4} - T_2^{\,4}) \qquad (2)$$

with
$\mathcal{F}_{1-2}$: view factor (explained below).
The last three considerations complicate the problem considerably. Sometimes these non-ideal factors are included in a transfer factor $\mathcal{F}_{1-2}$:

$$Q_{net} = A_1\mathcal{F}_{1-2}\sigma(T_1^{\,4} - T_2^{\,4}) \qquad (3)$$

4.2 View factors

Since radiation travels in straight lines, the amount of radiant energy transferred depends on "view factors" that take into account the area that one object can "see" in its geometric relation to another. For example, a disc shaped object appears different depending on the viewing angle. Fig. 3 [1] shows an example of this.

4.3 Wavelength dependence of emissivity

Another complicating factor in predicting radiant heat transfer from one surface to

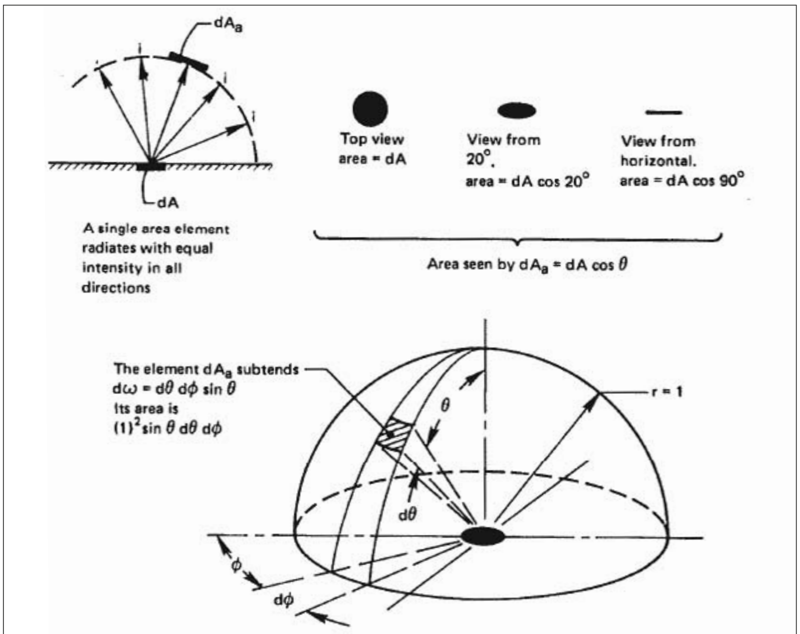

Fig. 3 **Radiation intensity through unit sphere**

another is the wavelength dependence of emissivity in real objects. In many cases, a "grey body" approximation can be used to simplify analysis. A grey body is one that has an emissivity that is not dependent on wavelength. Its emissive power is a constant fraction of the emissive power of a black body at the same temperature. No real body is grey, but many approximate grey behaviour. Fig. 4 [1] illustrates this for solar radiation as seen through the earth's atmosphere.

4.4 Specular and diffuse emission and reflection

Thermal energy emitted by a non-black surface, together with that portion of incoming radiant energy that is reflected from the surface, may leave the surface diffusely or specularly. A mirror reflects visible radiation almost perfectly specularly. When reflection and emission are diffuse, there is no preferred direction for outgoing rays. Black body emissions are always diffuse.

4.5 Simplifying assumptions used in analysis of the experiment

The approximation of diffuse grey bodies is used in the analysis. In this case an electrical analogy can be used. There are two new quantities used in this approach:

H [W/m²]: irradiance = flux of energy that irradiates a surface

B [W/m²]: radiosity = total flux of energy away from the surface

Radiosity can be expressed as the sum of the energy that is reflected by the surface and radiation emitted by it:

$$B = \rho H + \varepsilon e_b \quad (4)$$

which can be rearranged for H in terms of B:

$$H = \frac{B - \varepsilon e_b}{\rho} \quad (5)$$

with,

ρ: reflectivity

ε: emissivity

e_b: radiative power of a black body

The net heat flux per unit area (q_{net}) leaving any surface can be written as the difference of B and H for that surface is:

$$q_{net} = B - H \quad (6)$$

Substituting the equation 5 for H in terms of B we arrive at:

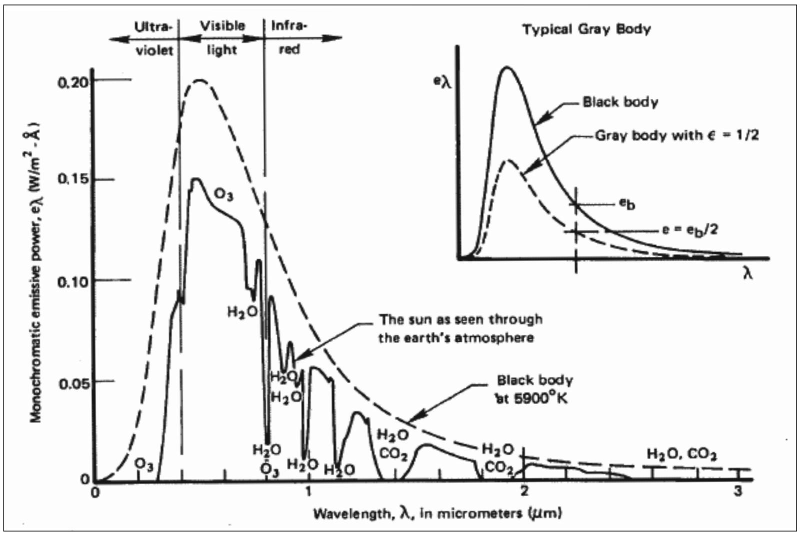

Fig. 4 **Comparison of the sun's energy as typicallly seen through the earth's atmosphere with that of a black body having the same mean temperature, size and distance from the earth**

$$q_{net} = B - \frac{B - \varepsilon e_b}{\rho} \quad (7)$$

Which can be rearranged to:

$$q_{net} = \frac{\varepsilon}{\rho} e_b - \frac{1-\rho}{\rho} B \quad (8)$$

If the surface is opaque, transmittance is zero, so 1 – ρ = α (where α is absorptivity), and if it is grey, α = ε.

From this results:

$$q_{net} A = Q_{net} = \frac{e_b - B}{\left(\frac{1-\varepsilon}{\varepsilon A}\right)} \quad (9)$$

This is analogous to Ohm's law, which tells us that (e_b–B) can be viewed as a driving potential (like voltage) for transferring heat away from a surface through an effective surface resistance,

(1–ε)/εA.

Now the heat transfer from one infinite grey plate to another parallel to it is considered as shown in Fig. 5 [1].

Fig. 5 shows a dotted line close to each surface that represents an imaginary surface that radiation flows past. If the grey plate is diffuse, its radiation has the same geometric distribution as that from a black body, and it will travel to other objects in the same way that black body radiation would. So, one can treat the radiation leaving the imaginary surface (the radiosity, B) as if it were black body radiation to another imaginary surface above the second plate. By analogy with the general form of the Stefan-Boltzmann (2) we have:

$$Q_{net1-2} = A_1 F_{1-2} \sigma (T_1^4 - T_2^4) = A_1 F_{1-2}(B_1 - B_2) = \frac{(B_1 - B_2)}{\left(\frac{1}{A_1 F_{1-2}}\right)} \quad (10)$$

The final fraction shows that this is also a form of Ohm's law. The radiosity difference can be said to drive heat through the geometrical resistance, $1/(A_1F_{1-2})$, which de-

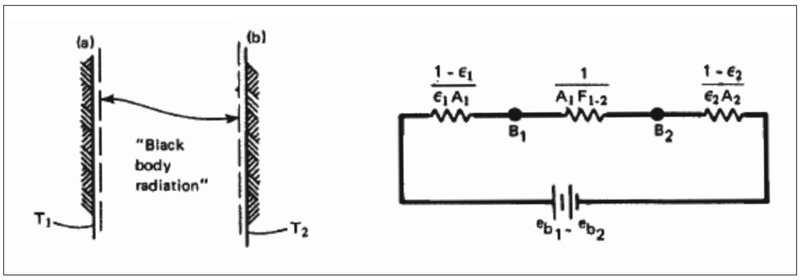

Fig. 5 **The electrical circuit analogy for radiation between two grey plates**

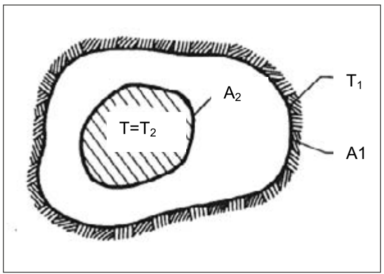

Fig. 6 **Heat transfer between an enclosed body and the body surrounding it**

scribes the field of view between the two surfaces. When two grey surfaces exchange heat only with each other, the net radiation flows through a surface resistance for each surface and a geometric resistance for the configuration. The electrical circuit shown in Fig. 5 shows this and allows to calculate Q_{net1-2} from Ohm's law. Recalling that $e_b = \sigma T^4$ (emissive power of a black body), can be obtained from:

$$Q_{net1-2} = \frac{e_{b1} - e_{b2}}{\sum resistances} = \frac{\sigma(T_1^4 - T_2^4)}{\left(\frac{1-\varepsilon_1}{\varepsilon_1 A_1}\right) + \frac{1}{A_1 F_{1-2}} + \left(\frac{1-\varepsilon_2}{\varepsilon_2 A_2}\right)} \quad (11)$$

For the particular case of infinite parallel plates, $F_{1-2} = 1$ and $A_1 = A_2$ it results:

$$Q_{net1-2} = A_1 \frac{1}{\left(\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1\right)} \sigma(T_1^4 - T_2^4) \quad (12)$$

Therefore, recalling (3) one can see that the transfer factor for infinite plates is:

$$\mathcal{F}_{1-2} = \frac{1}{\left(\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1\right)} \quad (13)$$

It also turns out that the electrical circuit analogy for one grey body enclosed in another (see Fig. 6) is exactly the same as for the case of two infinite parallel plates, except that in this case the areas are different. The equation for calculating the net heat transfer in this case becomes:

$$Q_{net1-2} = A_1 \underbrace{\frac{1}{\left(\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)\right)}}_{= \mathcal{F}_{1-2}} \sigma(T_1^4 - T_2^4) \quad (14)$$

4.6 Calculations of radiant heat transfer for the steel block experiment

The enclosed body case seems to be a reasonable approximation of the experiment of the block in a furnace. Now a fairly simple way to calculate Q_{net} for the heating of the block is given. For the interior areas of the furnace and block was measured:

Furnace area: 0,56 m²

Block area: 0,08 m²

Using our measured value for the emissivity of insulating firebrick (furnace lining) and a literature value for steel with an oxidized surface it results:

$\varepsilon_{Furnace}$: 0,49

ε_{Steel}: 0,80

At the moment the steel block is inserted into the furnace its temperature would be about 294 K (21 °C/70 °F). The furnace would be at 1366 K (1093 °C/2000 °F). The initial Q_{net} from (14) would be: Q_{net} = 11 270 W. If the furnace wall emissivity increased to 0,89 (as measured for our SiC coating) the initial Q_{net} would be: Q_{net} = 12 452 W.

This is a 10,5 % increase, not nearly as dramatic as the case discussed earlier for radiant transfer to space, but still significant. Increasing the emissivity of the space vehicle from 0,5 to 0,9 increased the heat transfer 1,8 times (80 % increase).

This calculation only tells us what the rate would be immediately after the block is placed in the furnace. As soon as its temperature increases incrementally, the temperature difference ($T_{Furnace}$–T_{Block}) would be smaller and the heat input calculated with

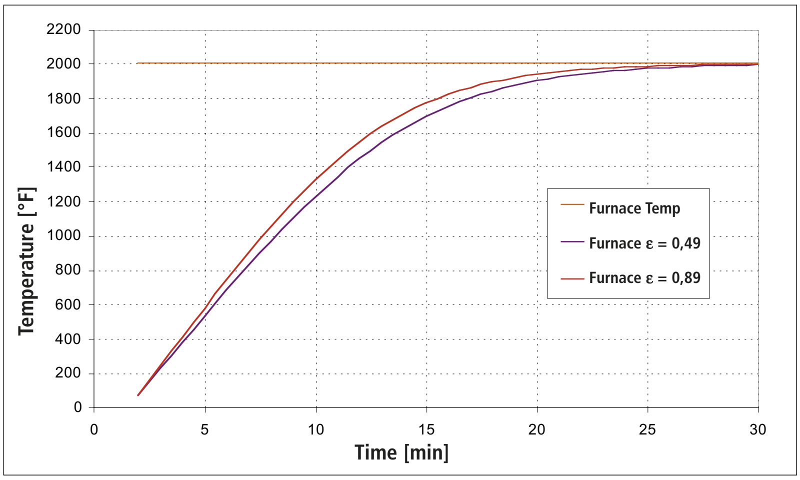

Fig. 7 **Profile of essential penetration phases from slag components into the alumosilicate-carbon composition (normalized relative phase quantities) without consideration of the contained carbon**

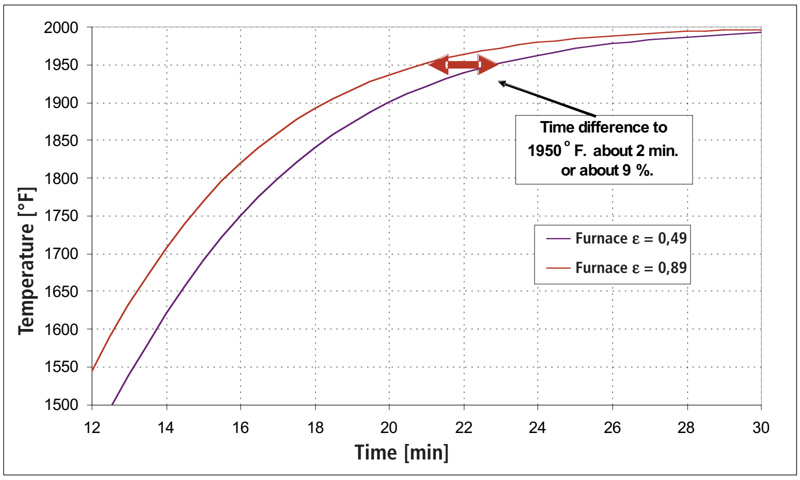

Fig. 8 **Heat-up rates for seel blocks based on theoretical calculations for radiant heat transfer only (1500 °F and above)**

equation 14 would decrease. The amount of heat required to increase the temperature of the block is given by its heat capacity and the mass of the block (volume × density). The values used for the steel block are as follows.

Heat capacity of steel: C_s = 490 J/kg·K

Density of steel: ρ_s = 7850 kg/m³

Volume of block: V_B = 0,00205 m³

The rate of temperature rise due to radiant heat transfer can be expressed as follows:

$$\frac{dT}{dt} = \frac{A_F \mathcal{F}_{F-B} \sigma}{C_s \rho_s V_B}(T_F^4 - T_B^4) \quad (15)$$

with

A_F: area of the furnace wall

T_F: furnace temperature

T_B: steel block temperature

This equation would be difficult to solve analytically, so it was approximated by calculating the heat input to the block in the first second using equation 14, then using the heat capacity, density and volume values to calculate the temperature rise for that second. Then, T_B was increased by the rise for the calculation of the next one-second time period. This calculation was repeated for each second by using the magic of computer spreadsheet analysis until T_B approached T_F. The results for two different furnace emissivities are shown in Figs. 7 and 8. As expected, the higher emissivity furnace would heat the block faster.

The asymptotic nature of the heat-up curves pinpointing the time that they reach 2000 °F (1093 °C) is impossible, but comparing the time to get close or (1950 °F/1065 °C) in Fig. 8 shows the following differences:

$\varepsilon_{Furnace}$: = 0,5; 22 min

$\varepsilon_{Furnace}$: = 0,9; 20 min

This is a 9 % reduction in time for the higher emissivity.

4.7 Consideration of the non-radiant heat transfer

The above calculations show that in a vacuum where the only mode of heat transfer is radiation, the block would be expected to heat-up ~9 % faster with the higher emissivity furnace wall. Obviously, radiation is not the only mode of heat transfer. We should be able to neglect heat conduction to the block since it is sitting on "skid rails" (ceramic cylinders), so it has minimal thermal contact with the bottom of the furnace. That leaves convection. Convective heat transfer can be expressed as follows.

$$Q_{net} = \bar{h}A(T_{fluid} - T_{body}) \quad (16)$$

with:

$\bar{h}$: average convective heat transfer coefficient over the body, W/m²·K

A: surface area of the body

As can be seen, this convective heat transfer is also dependent on a temperature difference, this time between the fluid (hot furnace gas) and the body (steel block). We will assume that the furnace gas and the walls are at the same temperature (1366 K) for the analysis. The block will receive energy from both radiation and convection. The combined net heat transfer is from adding the radiant contribution (14) and the convective (16):

$$Q_{net\ combined} = \bar{h}A_B(T_F - T_B) + A_F \mathcal{F}_{F-B}\sigma(T_F^4 - T_B^4) \quad (17)$$

Using the heat capacity, density and volume of the steel block we can calculate the temperature rise from the Q_{net} combined as follows:

$$\frac{dT}{dt} = \frac{\bar{h}A_B}{C_s \rho_s V_B}(T_F - T_B) + \frac{A_F \mathcal{F}_{F-B}\sigma}{C_s \rho_s V_B}(T_F^4 - T_B^4) \quad (18)$$

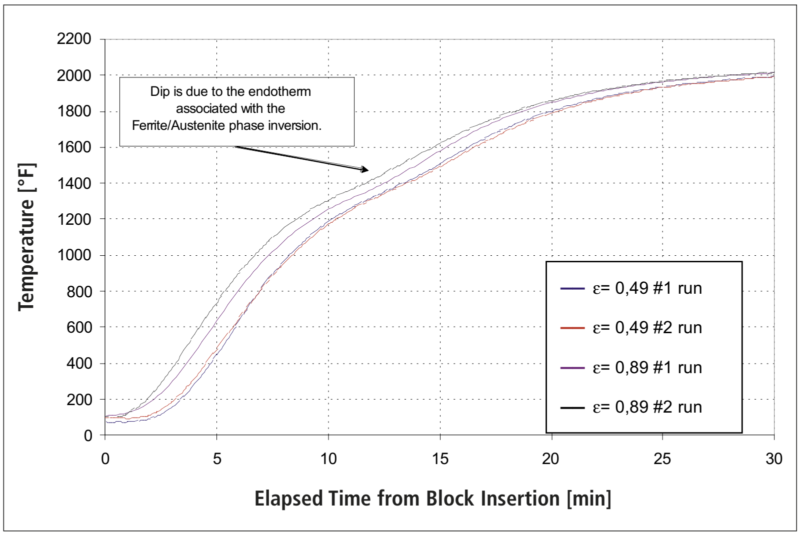

Fig. 9 Experimental steel block heat-up rates

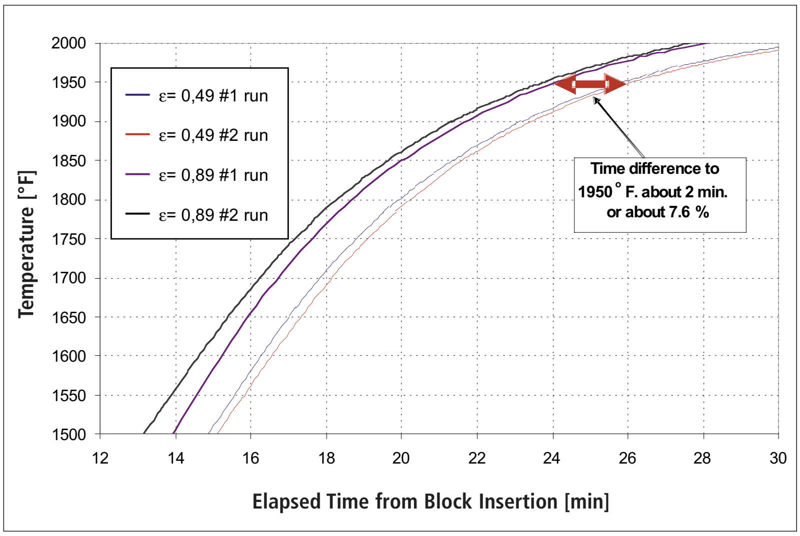

Fig. 10 Experimental steel block heat-up rates (1500 °F and above)

Once again, this equation would be difficult to solve analytically not to mention the fact; unlike the emissivity values we have no value for the convective heat transfer coefficient. Additionally we cannot assume 100 % efficiency in any of these heat transfer processes. Our only recourse at this point in the study is to run the experiment leaving only the emissivity of the lining as a variable and experimentally measure the difference.

Fig. 9 shows the heat up rates associated with 2 runs with the bare insulating brick of a measured emissivity 0,49 and 2 runs in the same furnace coated with a high emissivity coating measured as 0,89. Note the dip in the curve associated with the ferrite-austenite phase inversion giving us high confidence in the sensitivity of the experimental method. Fig. 10 is a graph showing only the 1500 °F range and above temperatures for clarity.

The improvement in heat up rate for the high emissivity coating closely matches the theoretically calculated 2 min but the overall heating rate is slower than calculated for radiation alone despite the fact that convective heating is clearly a significant part of the overall heat transfer. Currently the authors are studying what mechanisms may account for this. This has proven to be an experiment rich in information and as of this writing has not been fully analyzed.

5 Conclusions

- Theoretical calculations are presented that can be easily used to calculate the theoretical change in heat up of a load with emissivity changes to a furnace lining.
- Using these calculations the theoretical heat up rate of a 5-inch cube steel block was calculated and compared to actual experimental data.

Reference

[1] Lienhard IV., J.H.; Lienhard V., J.H.: A Heat Transfer Textbook. 3rd Ed. Cambridge, MA, 2008