A SENSOR FOR CONTINUOUS TEMPERATURE MEASUREMENT IN STEEL MAKING ENVIRONMENTS

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Abstract: Continuous temperature measurements are an integral part of most processes. They normally form a part of the control strategies for such processes. There is, however, a class of processes and, in particular, high temperature metallurgical processes where direct continuous temperature measurements are not easily possible. Such processes include steel making and copper making for example. Given that the output from steel making is approaching one billion tons of annual production on a global basis, one can see that being able to have continuous temperature measurements in several key units of the steel making process would be a significant development that would have an appreciable positive impact on quality and productivity.

A temperature sensor system consisting of a heat pipe is proposed to predict continuously, under extreme high temperature, high heat flux, environments, the liquid steel bath temperature. A heat pipe is a 'super-conducting' heat transfer device that uses the vaporization and condensation of a fluid to move energy from the evaporator to the condenser of the device.

A one-dimensional thermal analysis on the basis of thermal resistances is presented. It is shown that this analysis is capable of providing a correlation for predicting the liquid steel temperature as a function of some experimentally measurable temperatures such as the heat pipe internal temperature, the protective tube inner surface temperature and the air gap temperature. Also, sample experimental results including preliminary tests of an industrial trial are presented to show the potential of a McGill Heat Pipe, forming the core of the proposed sensor system for continuously measuring the temperature, as a reliable commercial sensor for measuring the liquid steel temperature in, for example, the tundish for the continuous casting of steel.

1. Introduction: The temperature in the tundish for the continuous casting of the steel, can be measured in two ways. The first is the manual immersion of disposable thermocouples. This method is reliable and it is used in many caster installations [1]. But, its' main disadvantages are that it is costly and if not exercised at sufficient frequency, some rapid temperature changes may go unnoticed. Unexpected increases in steel temperature can result in breakouts, and rapid temperature losses can lead to freeze-off and cast terminations [2]. Because of the demand for stable caster operation and reduced labor costs, continuous temperature measurement systems have been developed [1-3]. In one of the systems, the sensor used is comprised of a Pt-Rh/Pt-Rh thermocouple embedded in a refractory tube and it is immersed into the steel bath through an opening in the tundish cover. This arrangement does not benefit from a cooling system to protect the sensor and hence it has a limited life span. The application of a heat pipe – an extremely efficient heat transfer device – as a feasible temperature probe (sensor) has been previously studied by the authors [4]. The temperature probe consists of two heat pipes which are offset by a known length and the probe is not embedded in a refractory tube.

In the present paper, a model encompassing a temperature sensor system consisting of a heat pipe is proposed. Also, sample experimental results including preliminary tests of an industrial trial are presented to show the potential of a heat pipe as a reliable commercial temperature sensor.

2. Heat pipe concept: Generally, a heat pipe is a heat transfer device comprising two main sections: an evaporator and a condenser. It utilizes the vaporization and condensation of a working substance (water, sodium etc.) contained within to transfer heat from the evaporator to the condenser [5]. It is, in effect, a 'super-conductor' of heat energy. Fig. 1 illustrates a typical vertical heat pipe. During operation, heat is introduced to the sealed pipe shell from the heat source. This causes the working substance to evaporate. The vapor flows to the heat sink section where it condenses on the pipe wall and it returns to the evaporator by gravity in the liquid form.

On the basis of heat pipe technology, to handle high temperature and high heat flux situations, a new kind of heat transfer device named as a Thermopump or a McGill Heat Pipe has been developed [6]. Figures 2 and 3 show a schematic of a typical Thermopump and its' application as a temperature sensor in a furnace, respectively. The reader is referred to Reference [6] for a detailed account of Thermopump principles and its' distinct differences with a classical heat pipe.

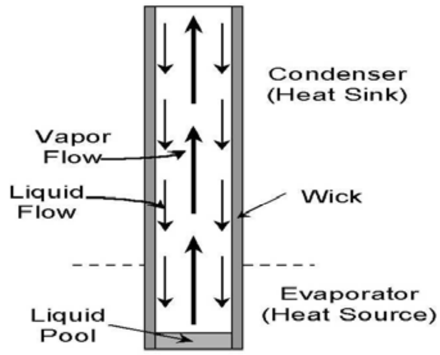
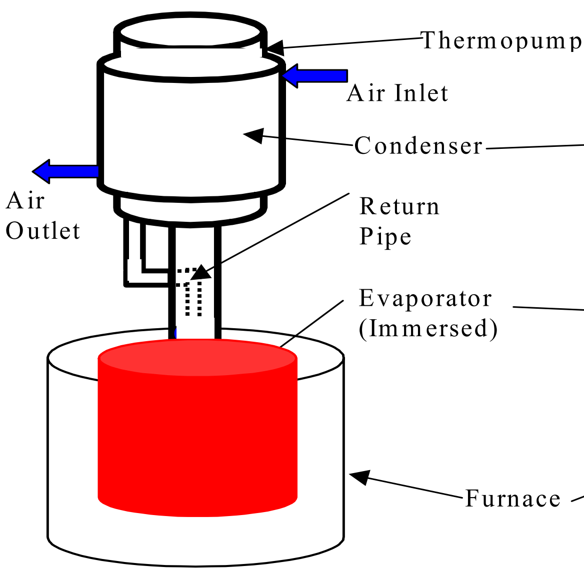


Figure 1: A typical vertical heat pipe

3. Experimental studies:

3.1 High temperature tests: In order to test the ability of a Thermopump as a high temperature sensor, a fixed length (7 inches) of its' evaporator section has been placed into a furnace heated directly by two natural gas burners. This arrangement is shown in Figures 2 and 3. The maximum temperature in the furnace is estimated as 1700 °C when both burners are set at maximum. Figure 4 represents the experimental results obtained with a water-based, air cooled Thermopump. That is, water as its' working substance and air as the fluid to cool the working substance in the condenser section. It can be seen that an experimental correlation, based on the sensor (Thermopump) internal fluid temperature, $T_{H.P}$, is employed to estimate the corresponding furnace environment temperature. The cooling air flow rate has been kept constant throughout the experiments.



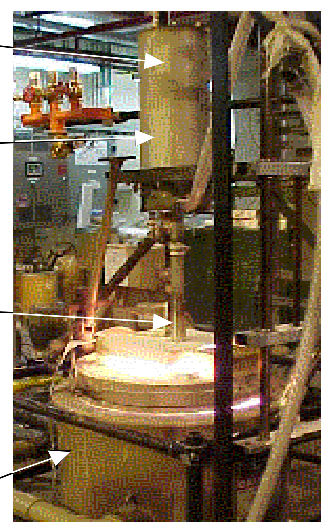


Figure 2: A schematic of a Thermopump   Figure 3: Thermopump as a temperature sensor

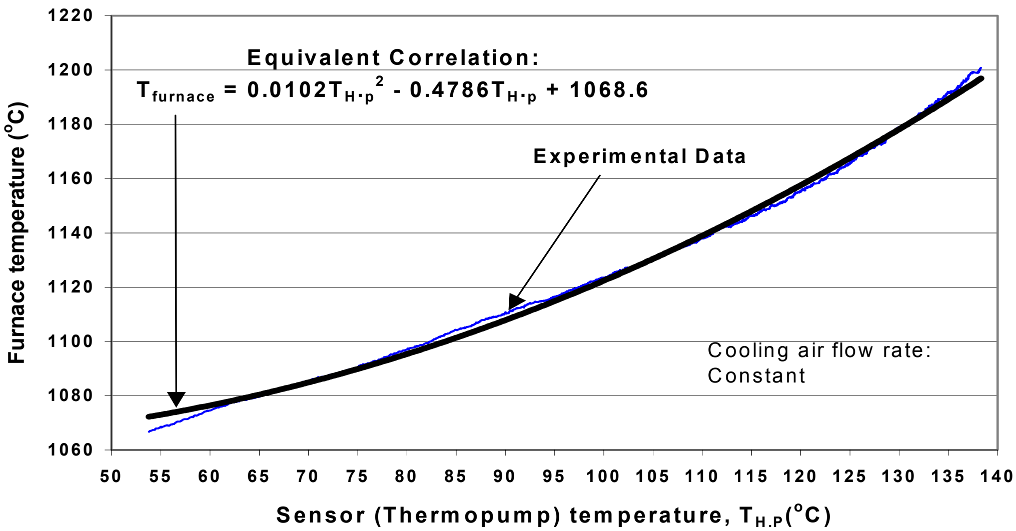


Figure 4: A Thermopump as a temperature sensor in a gas furnace

In addition, several high temperature tests as an industrial trial at RSR Technologies have recently been carried out. Figure 5 illustrates a schematic of the test set up. Lead acid batteries are recycled by being fed directly into a molten lead bath. A fixed length (26 inches beyond inner wall) of the evaporator section of a sodium-based, oxygen cooled lance (Thermopump) has been put into a furnace heated directly by two natural gas/oxygen burners. The furnace environment (freeboard) temperature is initially set at 3200 °F (1760 °C). Figure 6 shows some preliminary results indicating two important points. Firstly, the lance can withstand a very high furnace environment temperature due to being cooled by oxygen before O_2 is blown into a slag layer. Secondly, the lance, as a temperature sensor, correlates well with the furnace environment temperature because the environment temperature correlates with the feed rate. That is, by increasing the feed rate the furnace temperature is reduced and, in turn, the corresponding lance temperature is decreased. This general trend is demonstrated in Figure 6.

The gas/oxygen flow rate to both burners has been kept fixed during the tests with an exception that it is reduced by 5% at time stage of t_0. Also, the cooling oxygen flow rate has been kept constant.

It should be noticed that while sodium as the working substance of heat pipe is not a necessity, it is considered as a feature adopted for extreme environments as those encountered in steel making applications.

Subsequent tests will follow at RSR Technologies in the future, aiming to provide a better understanding of the capabilities of our lance (Thermopump) at elevated temperatures. This will open the way for carrying out future tests as industrial trials in steel making environments.

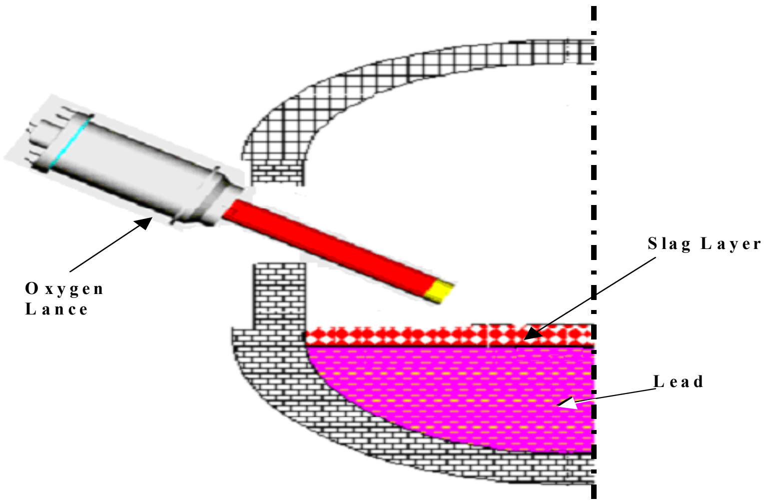


Figure 5: A schematic of an oxygen lance (Thermopump) in a lead recycling furnace

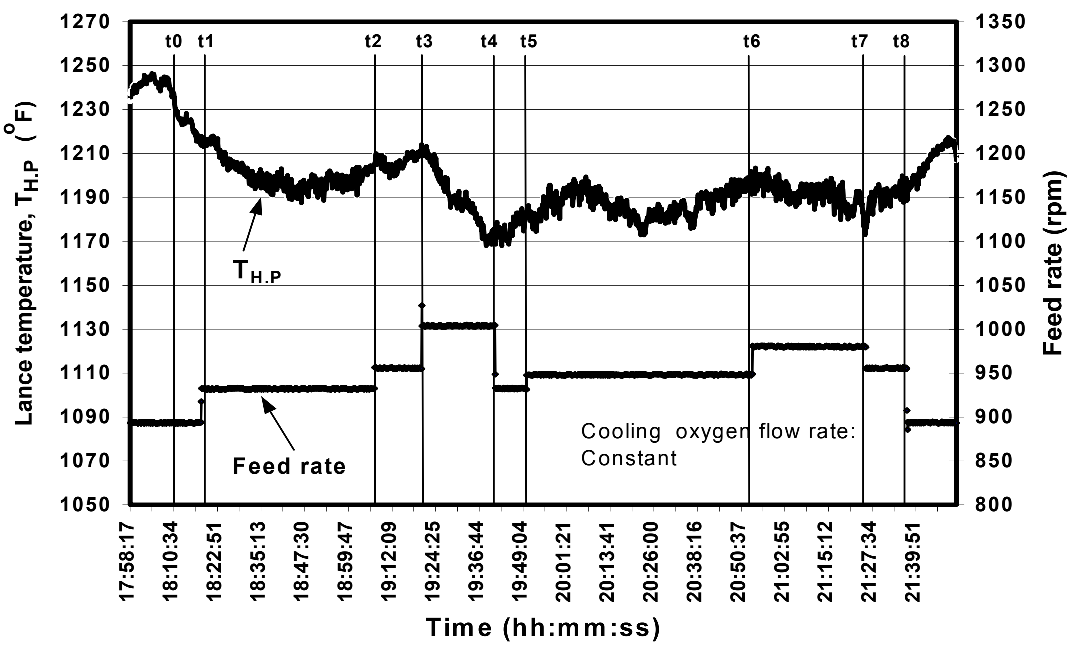


Figure 6: An oxygen lance as a temperature sensor in a lead recycling furnace

3.2 High heat flux tests: Further tests have been carried out to evaluate the ability of a Thermopump as a temperature sensor in a high heat flux environment. Hence, a fixed length (4.5 inches) of the evaporator section has been immersed into molten pure aluminum (about 12 kg) held in a silicon carbide crucible heated in the same furnace as above. Figure 7 illustrates the experimental results concerning the same Thermopump as in the previous section, except that water is used to cool the working substance in the condenser section. It is shown that an experimental correlation, based on the cooling water outlet temperature, T_w, is employed to estimate the corresponding molten aluminum temperature.

Tests have shown that a Thermopump could work under a high heat flux of 2.0 MW/m^2, which is over ten times higher than the critical heat flux of a classical water-based heat pipe [6].

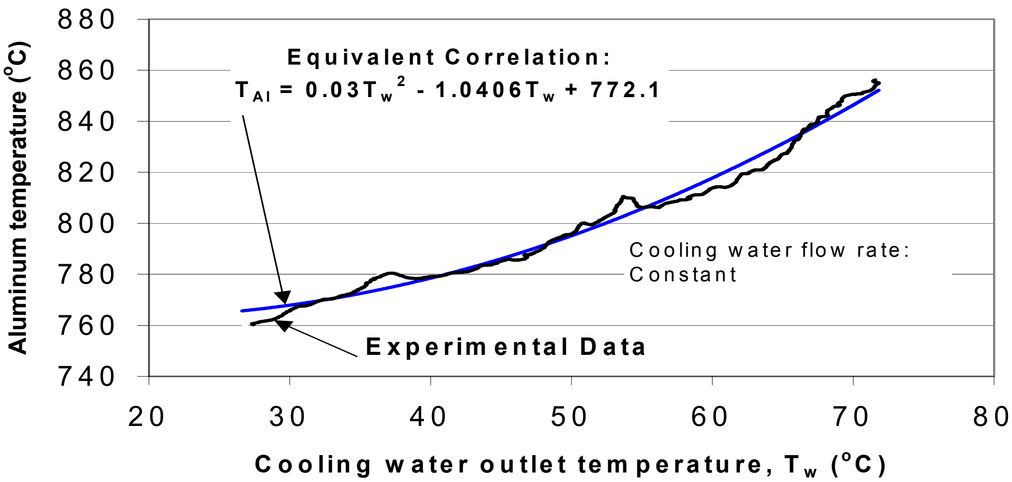


Figure 7: A Thermopump as a temperature sensor of molten aluminum

4. Temperature sensor system: In this section, a temperature sensor system consisting of a heat pipe (Thermopump) is proposed to predict continuously the liquid steel bath temperature. A one-dimensional thermal analysis on the basis of thermal resistances, is presented. It will be shown that this analysis is capable of providing a correlation for predicting the liquid steel temperature as a function of some experimentally measurable temperatures.

Figure 8 represents a schematic of the present temperature sensor system where a Thermopump forms the core. Also, a refractory tube is considered as an additional feature aiming to further protect the system in extremely high temperature, high heat flux, erosive and corrosive environments such as those encountered in, for example, the tundish for the continuous casting of steel. It is not, however, regarded as a necessity. Figure 9 shows the equivalent thermal circuit to the system in Figure 8, which is based on the steady-state and one-dimensional conditions. A one-dimensional heat transfer is assumed since the axial and circumferential heat conduction are negligible as compared with the radial one.

The thermal resistances shown in Figure 9 are described as follows:

$$\text{protective tube, } R_{t,cond.} = \frac{\ln(r_{o,2}/r_{i,2})}{2\pi L k_2} \quad \text{(K/W)} \tag{1}$$

Equation (1) is the thermal resistance for radial conduction in hollow cylinders [7]. $r_{o,2}$ and $r_{i,2}$ represent the outer and inner radius of the protective tube, respectively. The length, L, is considered as unity and k_2 is the thermal conductivity of the tube material.

Equation (2) shows the thermal resistances for surface radiation and radial conduction through the sealed gap that act in parallel. The narrow gap in between the protective tube and the heat pipe contains air with no bulk motion.

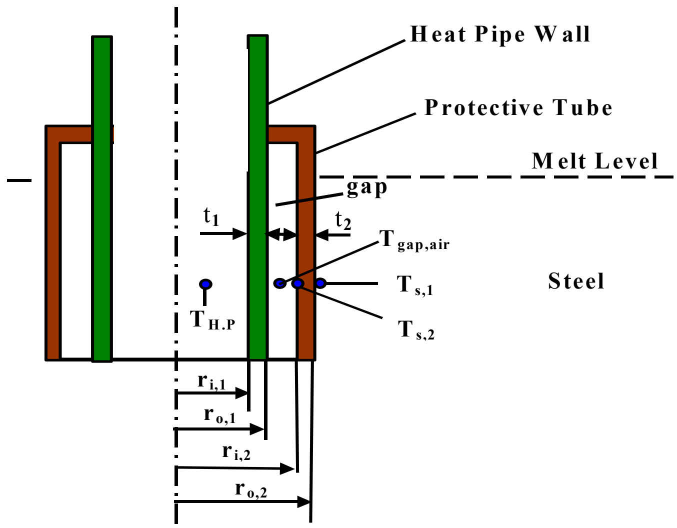


Figure 8: A schematic of temperature sensor system

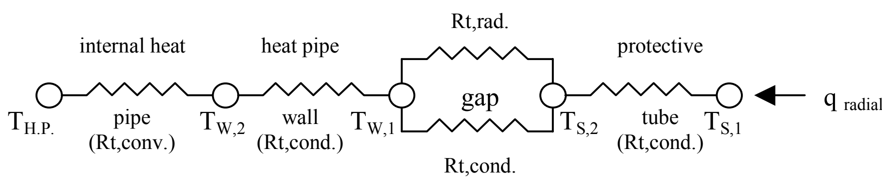


Figure 9: An equivalent thermal circuit for temperature sensor system

$$\text{gap(air)},\ R = \left[\frac{1}{R_{t,rad.}} + \frac{1}{R_{t,cond.}}\right]^{-1} = \left[\frac{1}{1/h_r.A_2} + \frac{1}{\ln(r_{i,2}/r_{o,1})/2\pi L k_{air}}\right]^{-1} \tag{2}$$

The effective heat transfer surface area, $A_2 = 2\pi r_{i,2} L$. Also, $r_{o,1}$ is the outer radius of the heat pipe and k_{air} is the thermal conductivity of air.

The radiation heat transfer coefficient, h_r, is obtained from equation (3) [7].

$$h_r = \varepsilon_{s,2}\sigma(T_{s,2} + T_{gap,air})(T_{s,2}{}^2 + T_{gap,air}{}^2) \qquad (\text{W/m}^2\text{K}) \tag{3}$$

It should be noted that the radiation mode has been modeled in a manner similar to convection. That is by linearizing the radiation rate equation, making the heat rate proportional to a temperature difference (i.e., $q_{rad} = h_r A (T_{s,2} - T_{gap,air})$) rather than to the difference between two temperatures to the fourth power. However, as illustrated in equation (3), h_r depends strongly on temperature (in Kelvin units). $\varepsilon_{s,2}$ is the emissivity of the protective tube material and σ is the Stefan-Boltzmann constant (= 5.670×10^{-8} W/m^{2}K^4).

$$\text{heat pipe wall},\ R_{t,cond.} = \frac{\ln(r_{o,1}/r_{i,1})}{2\pi L k_1} \tag{4}$$

In equation (4), $r_{o,1}$ and $r_{i,1}$ are the outer and inner radius of the heat pipe, consecutively. k_1 is the thermal conductivity of the heat pipe wall material.

$$\text{internal heat pipe, } R_{t,conv.} = \frac{1}{h_i.A_1} \tag{5}$$

Equation (5) represents the thermal resistance for internal forced convection with a phase change, referred to as two-phase flow, which is characterized by rapid changes from liquid to vapor in the flow direction. h_i is the two-phase heat transfer coefficient and $A_1 = 2\pi r_{i,1} L$.

Since the protective tube is in direct contact with the liquid steel bath, its' outer surface temperature, $T_{s,1}$, is considered as equal to the liquid steel temperature (see Figure 8). In order to obtain $T_{s,1}$, an energy balance is applied at the inner surface of the protective tube:

energy inflow, q_{in} = energy outflow, q_{out}

$$\frac{T_{s,1} - T_{s,2}}{\ln(r_{o,2}/r_{i,2})/2\pi L k_2} = \frac{T_{s,2} - T_{H.P}}{\sum R_t} \quad \text{(W)} \tag{6}$$

ΣR_t = summation of thermal resistances (= eq. (2) + eq. (4) + eq. (5)); $T_{s,2}$ = protective tube inner surface temperature; $T_{H.P}$ = heat pipe internal fluid temperature; $T_{gap,air}$ = air temperature in the gap. All of these temperatures can be measured by embedded thermocouples.

The following equation is obtained by rearranging equation (6).

$$T_{s,1} = T_{s,2} + \left[\frac{\ln(r_{o,2}/r_{i,2})}{2\pi L k_2}\right]\left(\frac{T_{s,2} - T_{H.P}}{\sum R_t}\right) \tag{7}$$

Equation (7) is employed as a correlation for predicting the liquid steel temperature, $T_{s,1}$, as a function of some experimentally measurable temperatures such as $T_{s,2}$, $T_{H.P}$ and $T_{gap,air}$.

In order to evaluate the accuracy of correlation equation (7) the influence of several parameters are investigated. This is done by carrying out some sample calculations on the basis of the following prescribed or assumed values:

k_1 (1000 K) = 25.4 (W/mK) (stainless steel, type 304 [7]); t_1 = 6 (mm); k_2 (room temperature) = 100 (W/mK) (graphite, medium grain, dense [8,9]); t_2 = 8 (mm); k_{air} (1250 K) = 0.08 (W/mK) [7]; $\varepsilon_{s,2}$ (0-3600 °C) = 0.75 (graphite, averaged [10]);

$r_{o,2} = r_{o,1}$ + gap + t_2 = 1 inch + ¼ inch + 8 mm = 39.75 (mm)

$r_{i,2} = r_{o,2} - t_2$ = 31.75 (mm)

$r_{i,1} = r_{o,1} - t_1$ = 19.4 (mm)

An initial estimation for $T_{s,2}$ is obtained by applying an energy balance at the outer surface of the protective tube:

$q''_{in} = q''_{out}$ (W/m^2)

Hence, $q''_r = \dfrac{k_2(T_{s,1} - T_{s,2})}{t_2}$ and therefore $T_{s,2} = T_{s,1} - \dfrac{q''_r \cdot t_2}{k_2}$ (8)

It should be noted that the practical temperature range in the tundish is 1500-1600 °C [1].

Influence of h_i: Typical values of h_i for convection with phase change (boiling or condensation) are in the range of 2500-100,000 W/m^{2}K [7]. Hence, for (a) h_i = 15,000 and (b) h_i = 70,000 and also assuming that $T_{s,2}$ = 1490 °C (for $T_{s,1}$ = 1530 °C and q''_r = 0.5 MW/m^2 in equation (8)), $T_{gap,air}$ = 1000 °C, and $T_{H.P}$ = 750 °C, the following results are estimated using correlation equation (7):

$T_{s,1}$(a) = 1515.7 °C and $T_{s,1}$(b) = 1516.8 °C

It can be seen that, firstly, the bigger h_i the smaller the thermal resistance $R_{t,conv.}$ (equation (5)) is and it results in a very small change in the value of predicted (estimated) $T_{s,1}$ (i.e., less than 0.1% deviation). Secondly, there is a nearly 1% difference between the predicted and prescribed values of $T_{s,1}$. This could well be improved by applying data from future experimental measurements for the above-assumed temperatures.

4.2 Influence of h_r: h_r is estimated for various values of $T_{gap,air}$ using equation (3). For (b) $T_{gap,air}$ = 1000 °C (as datum, previous section), (c) $T_{gap,air}$ = 800 °C (k_{air} (1050 K) = 0.07 W/mK [7]) and (d) $T_{gap,air}$ = 1200 °C (k_{air} (1450 K) = 0.092 W/mK [7]), and on the basis of previous assumptions, the following results are obtained employing equation (7):

$T_{s,1}$(b) = 1516.8 °C (h_r = 610.5 W/m^{2}K), $T_{s,1}$(c) = 1513.3 °C (h_r = 513.7 W/m^{2}K) and $T_{s,1}$(d) = 1520.8 °C (h_r = 726.3 W/m^{2}K)

It can be noticed that, firstly, the bigger h_r the more dominant the thermal resistance $R_{t,rad.}$ is as compared with the parallel resistance $R_{t,cond.}$ (equation (2)), and it causes a relatively small deviation of less than 0.25% in the value of predicted $T_{s,1}$. Secondly, the existing 1% difference between the predicted and prescribed (1530 °C) values of $T_{s,1}$ could be improved by using the future experimental data.

As a concluding remark, it is shown that equation (7) can be used as a flexible correlation to predict $T_{s,1}$. Within practical ranges, various parameters such as wall thickness, gap size and pipe diameter could be altered, as required, to fit the needs for a particular application.

5. Conclusions: (1) In the present work, a temperature sensor system consisting of a heat pipe (Thermopump) is proposed to predict continuously the liquid steel bath temperature. A one-dimensional thermal analysis on the basis of thermal resistances, is presented. It is shown that this analysis is capable of providing a correlation for predicting the liquid steel temperature as a function of some experimentally measurable temperatures.

(2) Sample experimental results including preliminary tests of an industrial trial, under high temperature and high heat flux conditions, are presented to show the potential of a McGill heat pipe (Thermopump), forming the core of the proposed system, as a reliable commercial temperature sensor.

6. References:

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