Study of heat transfer in tubular-panel and spray cooling systems applied to the electric arc furnace walls

A thesis presented by

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- I have acknowledged all main sources of help.
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___________________________
Josué Contreras Serna

Monterrey, Nuevo León, May 15th, 2018
Dedication

To my parents Joel and Teresa,
To my siblings Jorge, Vero and Joel.

To all my family and friends, who always believed in me.
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Abstract

This project consists in a heat transfer study in the electric arc furnace (EAF) walls, focused in the tubular-panel and spray cooling systems for the EAF located at Ternium-Guerrero plant, in the northeastern region of Mexico. The tubular-panel system is the one currently used to keep the walls cooled, composed of a total of 14 tubular panels. More dangerous accidents in the EAF operation, are the water steam explosions, which occur due to water leaks in the piping system inside the furnace. Spray cooling is given from the outside of the EAF, reducing the possibility of water directly impacting molten steel.

The main purpose of this research is to know the operational conditions of both systems, verifying if the spray cooling system could be as good as the tubular system for the removal of heat on walls, efficiency and keeping the walls at low temperatures.

The following procedures were used to estimate the water flow distribution in the cooling systems and the heat transfer in the walls. Piping network configurations are proposed for both systems. Models that consider surface-energy balances between different layers of the EAF's walls and the heat radiated onto the walls by the electric arc and the molten-slag surface are developed herein. Conventional correlations were used for determining the heat transfer coefficients inside the tubular panels (Internal convection) and alternate correlations for determining the heat transfer coefficients for the external convection (spray cooling). Additional scenarios were done trying to improve the operational conditions and heat removal of each system. Water flow regulation by valves in each panel in tubular system and jet nozzles are used instead of spray nozzles in the spray system to verify the effectiveness of the spray cooling. The study was conducted via a parametric analysis in which the
principal factors governing the process—the arc coverage and slag-layer thickness adhering to the walls—were varied.

The results of the tubular-panel system were compared with experimental measurements of the outlet water temperature in each panel, showing a good approximation; allowing us to predict the operational conditions of the furnace. For both systems the optimal operating condition of the EAF, is when the arc is completely covered and the maximal thickness of the slag-layer that can be reached is around to 4.5 cm, it does that energy losses to decrease significantly and to keep walls at low temperatures. The minimal temperature difference between the inlet and final flow is around to 3 K. The spray cooling system operates with a lower heat removal capacity and pressure than the tubular-panel, causing that inner wall surface temperature to be approximately 20 degrees higher than when using the tubular system for critical operating conditions. Under optimal operating conditions each nozzle removes approximately 8.5 kW of thermal power.

It is concluded that each cooling system has different temperatures and heat-removal capacity, which are highly dependent on the water flow within them. It is proved that slag-layer thickness and arc-coverage factors significantly affect the safe operation of the EAF, as well as its energy efficiency.

This is a semi-analytical study; the equations of models were obtained analytically, and an equation-solver program is necessary to treat the non-linear equations obtained. Relatively few computational resources are required for this method.
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Chapter 1. Introduction

Throughout history, steel has been of great importance for the development and growth of the world's population. Used as a main material, especially in the automotive industry, weapons industry, tooling industry, construction industry, among others. Only last year (2017), the total steel production in Mexico was about 20 million tons, 6.33% higher than 2016; while in USA a total of 81.6 million tons were produced and 13.70 million in Canada (Sánchez 2018). In México, Nuevo León is the third state with the highest steel production after Michoacán and Coahuila, however, the main corporations of the steel companies are concentrated in Nuevo León, such as Ternium, Deacero and Arcelor Mittal (Sánchez 2018).

The steel has been passed through out some processes until the decade of the 50’s, when the electric arc furnace (EAF) was established, most of first EAF uses alternate current (AC) and in the decades 90’s, the direct current (DC) EAF started to be developed by the interest of less energy consumption on these (Madias 2014).

Nowadays the EAF’s are one of the most important and used technologies around the world in the steel-making processes. The EAF machine is an impressive energy consumer, in some cases it can uses a power up to 150 MW without counting the chemical power, which could be of similar magnitude to the electric power! (Bowman and Krüger 2009).

As a comparison between the EAF’s and other big machines, in terms of power, Queen Mary 2, one of the biggest ships, and the Airbus A380, the passengers plane biggest in the world, both require a power around to 80 MW (Bowman and Krüger 2009); this tell us that the EAFs are very high consumers of energy. Therefore, the energy distribution inside the furnace is very important. Several studies have been
done trying to improve the steel-making processes and so, to reduce the energy losses and therefore the production costs of the steel.

The EAF studied in this thesis, is named “Danieli”, located in Ternium-Guerrero plant, northeastern region of Mexico (shown in the figure 1-1). It is a very big energy consumer and most of the time the electric arc is operating around to the 140 MW of power, so is quite important to study its heat transferring to know how the energy is impacting in the wall temperatures and to quantify the losses by the cooling systems.

Figure 1-1. Electric arc furnace Danieli, Guerrero plant-Ternium (Courtesy granted by the company TERNIUM).
1.1 Background

1.1.1 Steel-making process using EAF

EAFs are versatile, can use either scrap or direct reduced iron (DRI) to produce steel. The furnace studied operates principally with 30% of scrap and 70% of DRI. Most EAFs use alternating current (AC); however, there also exist direct-current (DC) furnaces, fabricated with the intention of a lower energy consumption (Madias 2014). The EAF studied herein is a DC furnace.

The steelmaking process of the EAF studied, needs an electric arc as the primary source of energy and heat, which is generated by two electrodes. The electric arc is used to melt the scrap and the DRI inside the furnace, by its radiation, to produce the steel, also other secondary products are generated as the slag, CO$_2$ and others. The steel should get up to 1600 °C to be molten. The slag is not at all a waste material, in fact, has a big importance in the steel making process. The slag works as a protector of the panels/walls covering the arc, the slag (foaming slag) has a lower density than the steel and it is over the molten surface covering a part of the arc. The slag has the property of adhering to the walls avoiding them to remove heat that could melt the steel faster and also protecting them to not be exposed directly to the arc, the heat of the melting surface and other gasses in the EAF. In general, the slag has the property of being porous and its capacity to conduct heat (thermal conductivity) can change according to the porosity present, for this study a sample of slag was taken from the EAF and its thermal conductivity was measured with the “KD2 Pro” thermal analyzer, estimating an average thermal conductivity about to 2.4 W / m-K, which is the value used for this study.

The figure 1-2 shows the process of the melting steel, it begins melting the scrap and DRI until the temperature of steel is around to 1600 °C. The process of steel production in EAF, takes approximately forty-five minutes to be done, the time it
Chapter 1

takes to load and unload the final material (tap to tap time), with an amount of 145 tons of steel by each cycle.

The heat (energy) generated inside the EAF is impressive, the temperature inside the EAF is in the order of 1800 °C and the electric arc has temperatures around to 10,000 K (Bowman and Krüger 2009), that is why the using of cooling systems as the tubular panels is very important; to keep the walls cold and to avoid damages and premature maintenances.

**Figure 1-2. Melting stage inside the EAF (Kruger, 2009).**

1.1.2 EAF wall cooling systems

To protect the furnace body from the high temperatures, first EAFs used refractory brick liners, as they don’t melt but are tended to break when the furnaces started operating at higher capacities (Gharib Mombeni, Hajidavalloo, and Behbahani-Nejad 2016) and (Ferguson and Zsamboky 2017). The solution was to protect the EAF roofs and walls with a cooling system of tubular panels with a high-pressure water pumped
for cooling, as is shown in the figure 1-3. In regular operation of EAF, the life expected for the refractory material was up to 350 heats, and then with the tubular cooling systems, the refractory wall life increased at least twenty-five times more (Grageda 1987).

The tubular water cooling panels allow the individual replacement of panels with a minimum of downtime, thus reducing maintenance times and costs, and increasing steel production.

Although the cooling system was improved, the charging and discharging of the EAF is repeated over 15 times in a day, causing thermal stress, thermal fatigue and finally small cracks in the pipes, shown in figure 1-4 (Gharib Mombeni, Hajidavalloo, and Behbahani-Nejad 2016). These cracks open up when the surface is cooled down, the water at very high pressure and volume may enter inside the furnace very quickly. The water leaks into a hot furnace can cause very violent water steam explosions when water is encapsulated by the molten steel, placing the people and the machines at high danger.
As an alternative to avoid the cracks in the pipes and therefore the steam explosions, an external cooling was proposed, spray cooling system, to cool the EAF roofs and walls. Main components of the spray cooling systems are the inner shell, outer shell, nozzles and water supply system, shown in figure 1-5. The water is sprayed against the hot plate as the cooling element of the inner shell, the outer shell is used as container of the water and steam. The water should be sprayed uniformly to the plate, this is essential to improve the heat removal. Just like in tubular systems, it is essential that a slag-layer be formed on the exposed side, to protects the panels or plate and improving the efficiency of the EAF process (Jones, Bowman, and Lefrank 1998). The advantage versus the panels systems is that water is sprayed at atmospheric pressure from the external side of EAF, thus the possibility of water being able to enter inside the furnace and fall on the molten steel is diminished.
As shown in the figure 1-3, the cooling panels have the property that a solid slag-layer adheres to them, this layer protects the walls from the heat since it works as insulation due to its low thermal conductivity. The slag-layer adhered could not have a uniform thickness along the surface of each panel, it may vary according to the height of panel. The slag can stick on the walls and roofs because it is splashed during the process or when emptying the molten steel from the EAF. For the spray systems, the slag can also be adhered to the steel plate since elements known as “C-cup” retainers are added to retain a high percentage of the slag, as the retainers used in the figure 1-6, also protecting walls from the heat inside the EAF. Similar retainers can also be added on cooling panels.
1.2 Justification/Motivation

The development of this thesis arises with the need to know the capacity of spray cooling systems to remove heat from the walls of electric arc furnaces, as it is known the spray cooling systems are safer to avoid the steam explosions, but there is not known with certainty about its heat removal. This study is also focused on the current tubular-panel system of the EAF, analyzing which of the two systems is more convenient to use for the cooling.

It is intended to know the optimal conditions that both systems must have to improve the use of energy inside the furnace.

1.3 Objective

1.3.1 General objective

To carry out a heat transfer parametric study for the cooling systems of the electric arc furnace walls, spraying and tubular-panel systems. The main varying factors in
the EAF process are the arc coverage, slag-layer thickness and the flow of water in the cooling systems.

1.3.2 Specific objectives

To determine the water flow distribution for the piping network configurations of both cooling systems, in order to estimate the heat transfer coefficient of each system according to the water flow rate trough every element of these systems.

To determine the temperature values in the different layers of the EAF walls, as well as the energy losses by the cooling systems.

To carry out a comparison study of the tubular-panel and spray cooling systems to verify which system keeps lower pressure drops, total water flow rate, temperatures and energy losses (better energy efficiency).

To analyze the options that can be made to improve cooling and to avoid explosions in EAFs. For the tubular panels cooling system is intended to control the flow of the coldest panels to maintain a uniform temperature in the wall, in addition, to verify how the water temperature increases when the flow in each panel begins to decrease. For the spray cooling system, a comparison is made of how the performance of the cooling system would be affected if nozzles used are for jet impingement.

1.4 Thesis organization

The present research is organized as follows:

- Chapter 1: A brief introduction and background are present; the motivation, objectives and scope of the research are listed.
• Chapter 2: The fundamental concepts of Fluid Mechanics and Heat Transfer are present, as well as the main characteristics of the spray cooling. Related previous works and contributions of this research are present.

• Chapter 3: The methodology used in the tubular-panel cooling system is present, the configuration of the pipe-cooling system in the EAF is shown. The models to find the distribution of the water flow and heat transfer are present. The results obtained for the tubular-panel system are shown, flow distribution, temperatures and energy losses for each panel, as well as the different scenarios that the EAF can operates if the water flow is regulated in the panels. The results are validated with experimental data of the water temperatures for most of panels in the process.

• Chapter 4: The methodology used in the spray cooling system is present, the configuration of the spray nozzles arrays is proposed and from this, the models of the distribution of water flow that goes through each nozzle and heat transferred in each nozzle section are generated. The different results obtained for the spray cooling system are shown, as flow distribution, temperatures and energy losses. Jet impingement, an alternative type of cooling, is compared against the spray cooling showing its capacity of heat removal and limitations of the system.

• Chapter 5: The operation capacities of each system are exposed, comparing them to each other, the advantages and disadvantages of the implementation of each system are discussed according to the results obtained for the models of each system.

• Chapter 6: Conclusions and future recommendations of this thesis are present.
Chapter 2. Theoretical framework and literature review

This chapter shows a general overview of the theory, explaining the principal concepts and elements behind the heat transfer study in the EAF walls, followed by the previous studies that have been done for the EAFs and are related to this study.

2.1 General Characteristics

2.1.1 Internal Flow fundamentals

Pipes have a great application in the industrial world, these are used mainly in heating or cooling material and to transport a fluid from one destination to another. For this study, pipes have the purpose to transport a fluid for cooling the EAF walls. Pipes can have different elements such as valves, nozzles, expansions, contractions, changes in flow direction, as well as they can have different geometries, dimensions, fluid types, materials, among others; all these elements are necessary to analyze to know the values of mass flow, head losses and pressure drops that pipes have.

Most of the flows in practice are turbulent and it is convenient to work with average speeds and constant properties for average temperatures, this has been showing a good accuracy (Cengel and Cimbala 2013).

2.1.1.1 Losses in piping systems

Pressure loss

Pressure loss is also known as pressure drop and occurs due to the viscous effects that exist in the pipeline and are irreversible.
The pressure loss commonly is expressed for all types of fully developed flows (laminar or turbulent) as:

\[ \Delta P_L = f \frac{L \rho^* V_{avg}^2}{D} \]  

(2.1)

Where \( f \) represents the friction factor and \( \rho^* V_{avg}^2 / 2 \) the dynamic pressure.

**Head Loss**

The pressure losses also can be represented in terms of equivalent fluid column height, head loss \( h_L \), the head loss is the additional height that the fluid should to be raised to overcome the friction losses, obtained by:

\[ h_L = \frac{\Delta P_L}{\rho g} = f \frac{L V_{avg}^2}{D \rho g} \]  

(2.2)

The friction factor for fully developed flow in pipes depends on the Reynolds number and relative roughness of the pipe material, Colebrook et al. (1939) developed the following equation to find the friction factor for turbulent flows in pipes, known as the Colebrook equation:

\[ \frac{1}{f_{0.5}} = -2 \log_{10} \left( \frac{2.51}{Re^{0.5}} + \left( \frac{\varepsilon}{D^3} \right) \right) \]  

(2.3)

**Major losses and minor losses**

Major losses and minor losses are associated with the flow in piping networks, in a piping system the fluid passes through some curvatures, tees, inlets, exits, contractions, expansions, valves and other fittings as well as the length of the pipe itself which can be a rough material. All these factors cause losses in the system
due to the flow separation and mixing, occasioning a drop in the flow rate and pressure.

Minor losses are the ones which are added by the fittings in pipes, as valves or nozzles, usually expressed in terms of the loss coefficient $K_L$ and are determined experimentally by the manufacturers.

Once the loss coefficient of a component is known, the head loss for that component (Minor Loss) is calculated with:

$$h_L = K_L \frac{v_{avg}^2}{2g} \quad (2.4)$$

The major losses are the head losses the flow have along the length of the piping network and can be also represented with the equation (2.2).

The total head loss in a piping system is determined with:

$$h_{L, total} = \sum_i f_i \frac{L_i v_i^2}{D_i 2g} + \sum_j K_{L,j} \frac{v_j^2}{2g} \quad (2.5)$$

For an entire pipe system or section with a constant diameter, the total head loss is reduced to:

$$h_{L, total} = \left( f \frac{L}{D} + \sum K_L \right) \frac{v_{avg}^2}{2g} \quad (2.6)$$

In piping systems, the total head loss $h_L$ is present in the steady-flow energy equation and can be written in terms of heads:

$$\frac{P_1}{\rho \cdot g} + \alpha \cdot \frac{v_1^2}{2 \cdot g} + z_1 + h_{pump} = \frac{P_2}{\rho \cdot g} + \alpha \cdot \frac{v_2^2}{2 \cdot g} + z_2 + h_{turb} + h_L \quad (2.7)$$
The $h_{\text{pump}}$ is the head from the pump to the fluid, $h_{\text{turb}}$ the head extracted, $\alpha$ is correction factor with a value of 1.05 for turbulent flows.

### 2.1.1.2 Piping configurations

Most of the piping systems used in industrial processes and in fluid distribution, as the residential, have several pipes connected to each other with parallel and series connections.

For pipes connected in series, the flow rate remains constant in the entire system regardless the diameter (mass conservation principle) and the total head loss is the sum of the head losses in each individual pipe. Figure 2-1 describes the case of series pipes.

![Figure 2-1. Pipes in series (Cengel and Cimbala 2013).](image)

For the parallel pipes the total flow rate is the sum of flow rate in each individual pipe and the head loss must be the same for each individual pipe. Figure 2-2 describes the case of parallel pipes.
Commonly in the analysis of piping networks to determine the flow distribution, an equation solver of simultaneous nonlinear equations is required, Engineering Equation Solver (EES) is used in this research.

### 2.1.2 Heat transfer fundamentals

It can be said that heat transfer is the mechanism where heat is transferred from one system to another, always transferred from the higher temperature system to the one with lower temperature (second law of thermodynamics). So, to heat transfer exists there must be a temperature difference between the systems (Cengel and Ghajar 2014).

The heat can be transferred by 3 main mechanisms, which are: conduction, radiation and convection, present in the study of this thesis.

#### 2.1.2.1 Energy balances

Most of the systems with a mass flow ($\dot{m}$) input and an output are analyzed under the condition of steady state, it is known as steady-flow system and the energy balance is reduced to:

$$\dot{Q} = \dot{m}\Delta h = \dot{m}c_p\Delta T$$  \hspace{1cm} (2.8)
Where $\dot{Q}$ is the rate of heat transferred entering or leaving the system, $\Delta h$ is the change of enthalpy and $c_p$ is the heat capacity.

On the other side we have the surface energy balance, where the energy that enters is equal to the energy (heat) that comes out when it passes through a surface, valid for steady and transient conditions:

$$\dot{Q}_{in} = \dot{Q}_{out} \tag{2.9}$$

### 2.1.2.2 Conduction

Conduction is the way in which the energy is transferred from the particles with greater energy to those adjacent with less energy through interactions of the particles.

The rate of heat conduction is given for the next equation, called Fourier’s law:

$$\dot{Q}_{cond} = -kA \frac{dT}{dx} \tag{2.10}$$

Heat conduction depends on the geometry of system, material and temperature difference. $A$ is the normal area, $dT/dx$ is the temperature gradient and $k$ is the thermal conductivity of the material which is the capacity to conduct the heat.

### 2.1.2.3 Convection

Convection is the mechanism of heat transfer that is given between a solid surface and a fluid that is in motion and passes through it, involves the combined effects of conduction and fluid motion, the Newton’s law of cooling express the convection heat transfer in the next equation:
\[ Q_{\text{conv}} = hA_s(T_s - T_\infty) \]  

\( h \) is the convection heat transfer coefficient (W/m\(^2\)-K), \( A_s \) is the surface area, \( T_s \) is surface temperature and \( T_\infty \) is the fluid temperature. The heat transfer coefficient is not a property of the fluid, it depends of many variables of the flow as the surface geometries, external or internal forced convection, natural convection, velocity of the fluid, properties of the fluid, roughness of the material, laminar or turbulent flow, among others. To estimate the convection heat transfer, many correlations in the literature are found for the different types of convection that can occur.

Some of the correlations used and explained in the following Chapters 3 and 4, are the equations (2.12, 2.13). The first one (2.12), is proposed by Gnielinski (1976) modifying the Petukhov’s equation to find a convection coefficient for internal flows with a very large numbers of Prandtl (\( Pr \)) and Reynolds (\( Re \)), the number of Nusselt (\( Nu \)) has as characteristic length the diameter of the pipe. Mudawar and Valentine (1989) proposed the correlation (2.13), to determine the heat transfer coefficient of sprayed water, using full-cone spray nozzles in the single-phase regime; \( Nu \) depends on the Sauter mean diameter of the droplets as the characteristic length and the spray Reynolds number (\( Re_s \)) is in function of the volumetric flow over the surface impacted as the velocity.

\[
Nu = \frac{\left(Re - 1000\right) \cdot Pr_{water}}{1 + 12.7 \cdot \left(Re_s^{0.5} \cdot Pr_{water}^{0.5}\right)}
\]  

\[
Nu = 2.512 \times (Re_s^{0.76}) \times (Pr_{water}^{0.56})
\]  

2.1.2.4 Radiation

Radiation is the energy that is emitted in the form of electromagnetic waves from one surface to another, also known as surroundings, it is a volumetric phenomenon and can affects solids, liquids and gases. The radiation that is emitted by the bodies due to its temperature is known as thermal radiation, it can be expressed in the
following way when a surface is separated from a black surface that does not intervene with the radiation (radiation is not reflected):

$$\dot{Q}_{\text{rad}} = \varepsilon \sigma F A_s (T_s^4 - T_{\text{surr}}^4)$$ (2.14)

Where $\varepsilon$ is the emissivity of the surface and it is a property of the surface material, $\sigma$ is the Stefan-Boltzmann constant ($5.67 \times 10^{-8}$ W/m$^2$-K$^4$) and $F$ is the view factor that is the fraction of radiation that is streaking the receiving surface.

### 2.1.2.5 Thermal Resistance

Analog to the electrical resistance in an electrical circuit, there is the thermal resistance ($R$), which is the opposition that presents a system against heat flow, either by conduction, convection, radiation or a combination of these. The rate of heat transfer can be estimated by using thermal resistances, as shown in the next equation:

$$\dot{Q} = \frac{\Delta T}{R}$$ (2.15)

Where $\Delta T$ is difference of temperature between two surfaces and $R$ the thermal resistance (K/W), better represented in the figure 2-3 in the analogous way.

![Figure 2-3. Analogy thermal resistance and electrical resistance. A) Heat flow. B) Electric current flow (Cengel and Ghajar 2014).](image)
2.1.3 Spray cooling

The spray cooling is a technology that uses tiny droplets of a fluid liquid as the cooling medium impinging a heated surface, increasing the effectiveness of the heat transfer. Spray cooling has been used for several purposes as fire protection, cooling chips in electronic devices, biomedical industry, cooling of hot gases and also in the steel-making processes to cool surfaces that are up to 1800 K (Yan et al. 2012).

Another process similar to the spray cooling is the jet impingement, these are two-phase liquid cooling schemes (Liang and Mudawar 2017).

Both schemes are given by means of a nozzle, the nozzles for spray cause the flow to be broken into tiny droplets and the jet nozzles make the liquid come out in a compact stream jet with a defined jet length. Some applications of the jet impingement are cleaning, cutting and also for cooling systems. The figure 2-4 shows the physical differences that both systems have when the liquid impacts a surface.

The spray cooling presents several advantages against the jet impingement: high heat flux dissipation, lower and uniform surface temperatures and has the capacity to cool a larger surface area by using a single nozzle (Liang and Mudawar 2017).
The jet impingement can cause tendency to separate the jet from the impingement zone due the vapor perpendicular to the surface, with the spray cooling the vapor is removed easily from the heated surface, allowing more effective contact of the droplets and therefore for the heat removal (Liang and Mudawar 2017).

2.1.3.1 Nozzle types and orientations

There are two types of nozzles that are used to generate the spray, which are the atomized nozzles assisted and controlled with air or a gas to break the flow and create the small drops and; pressure nozzles where the spray is produced (small fine drops) by a high pressure and preferred for cooling purposes (Visaria and Mudawar 2009). The figure 2-5 shows the different forms of spray that can be obtained depending on the type of nozzle.

![Pressure Nozzle Diagram](image)

Figure 2-5. Droplet impact patterns of pressure nozzles (Liang and Mudawar 2017).
Hollow cone concentrates most of droplets in the circumference of the impact area, Flat spray is a narrow oval in the impact area and in the full cone pattern the spray covers the area of impact completely, that's why they are the ones that are preferred to be used for cooling.

The spray nozzles could have many orientations which are downward-facing, upward-facing, horizontal and inclined. Also, a minimal distance of the nozzles to the surface is required to produce the fully spray pattern. However Rybicki and Mudawar (2006) proved that orientation of the spray nozzle no represents a significant influence for cooling performance.

The inclination of the spray nozzles has some effects on heat transfer. Wang et al. (2010) showed that the heat transfer performance is enhanced when increasing the spray inclination angle. Hunnell et al. (2006) tried two types of spray cooling orientation (vertical and horizontal), finding that horizontal spray has a slightly better heat transfer removal than a vertical downward spray when using a flow rate of 9.8x10^{-6} m^3/s; and the performance for vertical and horizontal spray configurations is very similar for low flow rates. Bolle and Moureau (1976) showed minimal influence of the spray orientation, they suggested that the heat transfer effects are given principally by the initial impact of droplets and not the runoff flow on the surface.

2.1.3.2 Spray Boiling curve

The heat transfer in the spray cooling processes can be given within 4 different boiling regimes and can be represented with the next spray boiling curve (figure 2-6).
In the metal heat-treating, the cooling starts in the film boiling regime where the gradient of temperature is larger, a thin vapor film is over the surface avoiding the direct contact of the droplets and showing a poor heat transfer coefficient and heat removal. The minimum heat flux, also known as Leidenfrost point, marks the beginning where the insulating vapor film starts to break up, improving cooling rate and starting the transition boiling regime (Mascarenhas and Mudawar 2010). In the transition boiling regime, both nucleate and filming partially occur, the top of the curve, Critical Heat Flux, is the point where the heat flux reaches a maximum. Transition boiling regime is also known as “unstable film boiling regime” and is avoided in the practice (Cengel and Ghajar 2014). After the critical heat flux point,
the surface is available to be wet by the liquid droplets, also with high heat removal rates known as nucleate boiling regime where vapor bubbles are present. Finally the regime where there are no bubbles or a boiling formation present in the surface, is called the single-phase regime and the cooling rate decreases considerably (Mascarenhas and Mudawar 2010). In the single-phase regime the heat transfer is mainly given by the hydrodynamics of the spray.

Different correlations are found in the literature for the four spray boiling regimes, in order to determine the heat flux and convection heat transfer coefficients of the sprayed water.

2.1.3.3 Main parameters in spray cooling

The cooling performance of the spray cooling depends on many parameters, some of them are as follows:

- Type of fluid.
- Injection and ambient pressure.
- Surface parameters: surface temperature, heat flux and thermal conductivity.
- Flow parameters: flow rate and pressure drops across the nozzle.
- Nozzle parameters: orifice diameter of nozzle.

There are spray parameters with a crucial role like the droplet sizes and volumetric flow (Liang and Mudawar 2017).

In the spray cooling the droplets are formed with different diameters therefore is more convenient working with a mean droplet diameter. The most common used droplet diameter is the Sauter mean diameter \(d_{32}\) defined as the diameter where volume-to-surface area ratio is the same in the entire spray. Another diameter, is the mass median diameter \(d_{0.5}\), which the 50% of the total liquid volume is in the
droplets with smaller size (Liang and Mudawar 2017). In the literature different correlations to estimate the values of the droplet diameters can be found.

Mudawar and Valentine (1989) proved that the spray volumetric flux, volumetric flow over the surface area impacted by the sprayed water (m$^3$/s/m$^2$), is the most important parameter. It has the dominant effect on heat transfer than the other hydrodynamic parameters.

The heat transfer is increased whit increasing the flow. For spray cooling, a larger flow results in a higher velocity of the fluid over the surface with a thinner thermal boundary layer. The impact of droplets onto the film agitates the fluid, thinning the thermal boundary layer (Kim 2007).

2.1.3.4 How the spray pattern is obtained

The full cone spray in nozzles is obtained by swirls inside the nozzles created by spiral grooves milled in the nozzle chamber, the fast-swirling fluid experiments centrifugal forces capable to break the liquid sheet format into small droplets, which are tended to be of similar sizes and distributed uniformly in the spray area (Ashgriz 2011). In the figure 2-7 show a transversal cut of one nozzle typical used to cool the EAF walls and roofs. The figure 2-8 shows the typically action mechanism of the spray full cone – nozzles, case (A) for axial flow and (B) for tangential flow.
2.2 Previous work and contribution

2.2.1 Tubular-panel cooling system.

There are some important investigations that have been studied the heat-transfer phenomena in the EAFs, with similar purposes of understanding how the energy is being used in these complex machines and what are the principal characteristics that should be considered to help workers in the steel-making industry handling furnaces on a daily basis. For instance, E. Trejo et al. (2012) proposed a method to calculate the energy losses of an entire EAF (including losses of the cooling system with a value around of 10%) according to the energy per ton of steel (kWh/T). The
calculations were performed by proposing a heat-radiation model representing the resistive elements within an electrical network and finding the radiation view factors using a computational fluid dynamics (CFD) simulation. Trejo et al. also studied the effects of the slag thickness and arc coverage; however, the tubular panels not were analyzed in detail, the EAF walls were only split into two sections, and the flow distribution of water was not considered. Logar et al. (Logar, Dovzan, and Skrjanc 2012a, 2012b) estimated the heat transfer and temperatures of the entire EAF with a transient method, the arc power is considered in the analysis. However, the outlet water temperatures were not calculated, the wall was considered as one section, and the water flow was considered invariable, showing surface temperatures around 315 K and an estimation of 15% of energy losses in the cooling system. Khodabandeh et al. (2017) performed a numerical CFD simulation of an entire EAF, dividing the wall into two sections; they principally investigated the effects of furnace-geometrical characteristics and slag thickness upon the temperatures distribution inside the furnace. Mombeni et al. (2016) studied the heat transfer in the EAF roof-cooling pipes using transient simulation, obtaining an outlet water temperature of 330 K. They also found that the main points at which dramatic changes may lead to cracking in the future occur around the bends in the piping.

This study determines the temperatures of each individual panel on the wall, as well as the heat being removed (losses) by each one according to its water flow and geometries. This thesis shows which panels are the hottest and presents the safe and efficient operational conditions of an EAF. Thus, several measures can be implemented to improve the performance of the process and avoid inopportune maintenance due to leaks, cracks or damage to the hottest panels; these measures include controlling the flow to the panels with greatest water flow and lower temperatures, as well as by improving the conditions of slag adhesion and arc coverage. The slag is very helpful to the EAF process, as it improves the arc stability, and when coated on water-cooled panels, improves their thermal efficiency by reducing the heat losses using the cooling system (Bowman and Krüger 2009). The
slag adhered to the walls works as an isolation to avoid the walls to get heated up, as well as it protects them from the arc radiation.

### 2.2.2 Spray cooling system.

The spray cooling is one of the most interest areas of heat-transfer, some authors have investigated it in the single-phase regime to see its hydrodynamic characteristics for cooling purposes. Abbasi et al. (2010) predicted the local heat transfer according to the normal pressure exerted by the spray, determining the heat transfer coefficient (HTC) using different types of nozzles. Abbasi obtained a HTC around to 14.45 kW/m²-K using the correlation they proposed and 11.14 kW/m²-K with the experiment realized for a spray cone flow with a pressure of 206 kPa. Mzad and Khelif (2016), determine the heat flux and heat transfer coefficient of the spray for different spray pressures, founding the heat transfer coefficient is bigger when the pressure increments. The spray heat transfer coefficient they obtain, was in the range of 9x10³-39x10³ W/m²-K for surface temperatures from 100 – 815 C. Other studies were performed determining heat transfer coefficients and other spray characteristics, these are mentioned in the chapter 4 where the models of the spray cooling system are performed.

Ferguson and Zsamboky (2017) show the advantages of the spray cooling systems against the tubular-panel systems, claiming that one of the principal advantages is the safety operation, as the pressure used in these systems is almost the half of the tubular-panel pressure and it is cooling from the outside. They mention that the downtime of the EAF process are minimized as the cooling system maintenance times are lower, achieving the costs reduction. They present several steelmakers that are using the spray cooling systems around the world. Smith and Ward (2012) discuss the operation conditions of the EAF using a spray cooling system in the roofs. They show that the system used has a good capacity to retain and form slag-layer on the walls, using “C-cup” slag retainers prolonging the equipment life. They estimated some design calculations for an exposed surface area of 630 ft², obtaining
a heat flux of 30950 Btu/hr-ft² (around to 5.714 MW), these are the energy losses if the system they suggest is implemented.

This thesis aims to show the temperatures of the walls that could be obtained if the spray cooling systems were implemented in the EAF, as well as the amount of heat (losses) that these systems are removing when they are operating in the single-phase regime. The optimal conditions of the EAF operation, when using spray are shown; and a simple comparison against the jet impingement is performed to prove if spray is more reliable for cooling.

A comparative analysis of both systems (spray cooling and tubular panel cooling) in this study is performed, showing the similarities and differences of each system, which of them affects the efficiency, temperatures, safety and other operational conditions of EAFs.
Chapter 3. Tubular-panel cooling system

The models for the flow distribution of water inside the tubular panels and the heat transfer are presented. The values for the parameters used are listed in tables in the following sections.

3.1 Flow-distribution model

The EAF wall is composed of 14 tubular panels. A parallel-piping-network model is proposed, the schematic of this is shown in figure 3-1. Here the total initial volumetric flow ($\dot{V}_o$) is a known real value measured by sensors in the EAF. The Table 3-1 shows the list of the parameters used for the flow distribution model. Assumptions of this model are as follows:

- Steady state.
- Pressure losses are neglected for the flow path that feeds each panel.
- The internal diameter of pipes is constant, and the material of the pipes is the same for all lengths.
- Water properties are constant and taken at 315 K bulk temperature.
- The outlet water flow is equal to the inlet flow.
Figure 3-1. Parallel piping network considered in the flow distribution model.

The figure 3-2 shows more clearly that the tubular panels are connected in parallel, where the flow rate is injected to each panel from the top and the back wall of the EAF through a main pipe and leaves through a pipe at the bottom to reconnect with the flow from the other panels. Thus, the total outlet flow rate equals the inlet flow rate, meaning the total flow is the sum of the flows of all panels (3.1). As the network is parallel, all the head losses \( (h_L) \) in the panels piping are the same (3.2). The diameters of all pipes are constant, the head losses are calculated adding the major losses plus the minor losses \( (K_L) \) in the equation (3.3) (Cengel and Cimbala 2013). Major losses depend of the friction factor and the wet piping length of each panel \( (L_{pn}) \), the panels 6 and 12 are the longest and the panels 7, 8 and 9 are shorter. Minor losses consist of the loss coefficients of valves and curvatures \( (K_{LVpn}, K_{Lcpn}) \) equation (3.4), which were obtained according to the curvatures and bends of each panel piping. The panels 7, 8 and 9 have the lowest loss coefficient values (shown in Table 1). As shown in the figure 3-3 the panels have different geometries and configurations, the panels 1,4,5,7,8,9,10,11,14 have a similar configuration as the figure 3-3 (A), the panels 2, 6 and 12 have similar configuration to figure 3-3 (B); the panels 3 and 13 are like figure 3-3 (C). The images in figure 3-2 and 3-3 are not in scale and do not have the same dimensions and configurations as the real panels,
they are only used for representative purposes and the panels may have different surface areas and lengths, as shown in table 3-1.

\[ \dot{V}_o = \dot{V}_{P1} + \dot{V}_{P2} + \ldots + \dot{V}_{P14} \]  
\[ h_{Lp1} = h_{Lp2} = h_{Lp3} = \ldots = h_{Lp14} \]  
\[ h_{Lpn} = \left( \frac{f_{pn} L_{pn}}{d_{pipe}} + K_{Lpn} \right) \cdot \left( \frac{V_{pn}^2}{2 + g} \right) \]  
\[ K_{Lpn} = K_{LVpn} + K_{Lcpn} \]

*Figure 3-2. Distribution and paralleling connection of the tubular panels in the EAF.*
Figure 3-3. Geometry of the panels. A) Regular panel. B) Gas injection panel. C) Panel for sampling.

The friction factor ($f$) is obtained with Colebrook’s equation (3.5) (Colebrook et al. 1939), which depends on the Reynolds number ($Re$) and roughness ($e$) of piping of each panel.

$$\frac{1}{f_{pn}} = -2 \times \log_{10} \left( \left( \frac{2.51}{Re_{pn} \times f_{pn}} \right) + \left( \frac{e}{d_{pipe} \times 3.7} \right) \right)$$  \hspace{1cm} (3.5)

The equations (3.6, 3.7) determine mean water velocity ($V$) and Re in piping of each panel, which are necessary to solve the system equations. The same number of unknowns and equations are obtained.

$$V_{pn} = \frac{v_{pn}}{A_{intpipe}}$$  \hspace{1cm} (3.6)

$$Re_{pn} = \rho_{water} \times V_{pn} \times \frac{d_{pipe}}{\mu_{water}}$$  \hspace{1cm} (3.7)
Chapter 3

Table 3-1. Parameters used in the flow distribution model and dimensions of tubular panels.

<table>
<thead>
<tr>
<th>Panel</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{pn}$ (m)</td>
<td>47.16</td>
<td>37.55</td>
<td>35.39</td>
<td>37.18</td>
<td>37.18</td>
<td>39.42</td>
<td>26.64</td>
<td>27.19</td>
<td>29.66</td>
<td>37.18</td>
<td>37.18</td>
<td>39.42</td>
<td>35.39</td>
<td>37.18</td>
</tr>
<tr>
<td>$K_{Lpn}$</td>
<td>9.2</td>
<td>10.7</td>
<td>7.1</td>
<td>6</td>
<td>6</td>
<td>12</td>
<td>2.8</td>
<td>2.8</td>
<td>2.8</td>
<td>6</td>
<td>6</td>
<td>12</td>
<td>7.1</td>
<td>6</td>
</tr>
<tr>
<td>$A_{pn}$ (m$^2$)</td>
<td>0.045Aw</td>
<td>0.062A</td>
<td>0.062A</td>
<td>0.062A</td>
<td>0.062A</td>
<td>0.062A</td>
<td>0.046A</td>
<td>0.048A</td>
<td>0.058A</td>
<td>0.062A</td>
<td>0.062A</td>
<td>0.062A</td>
<td>0.062A</td>
<td>0.062A</td>
</tr>
<tr>
<td>w</td>
<td>w</td>
<td>w</td>
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<td>w</td>
<td>w</td>
<td>w</td>
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<td>w</td>
<td>w</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

3.2 Heat-transfer model

A heat transfer model is established for the EAF walls. The main considered assumptions are as follows:

- Steady state.
- The EAF is considered with a cylindrical shape.
- The heat transfer from the molten surface and electric arc to the walls, is only given by the radiation mechanism.
- The electric arc is considered as a cylinder and with a diameter of 0.15 m, Bowman and Krueger (Bowman and Krüger 2009) claims the shape of the arc as a cone, but it is taken as a cylinder to simplify the analysis of view factors.
- The walls are considered isolated from the environment, the heat received from the arc and molten surface is completely removed by the cooling system.
- The radiation from the electrodes and the convection heat of the gases are neglected.
- All the radiation transferred from the arc and molten surface is absorbed in the walls, it is not reflected.
- It is assumed the molten surface with a temperature of 1873 K (1600°C) and the electric arc has a temperature of 10000 K (Bowman and Krüger 2009).

Internal Pipes diameter = 0.0667 m  
Pipes Roughness = 0.045 E-3 m  
Loss coefficient of valves = 0.2  
Initial Volumetric Flow = 0.3194 m$^3$/s
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The figure 3-4 shows the elements used to model heat transfer in the EAF-walls study when using the tubular-panel system.

![Figure 3-4. Elements of the EAF used in the heat transfer model for the tubular-panel system.](image)

The model is represented by the three principal equations (3.8, 3.15 and 3.16). The equation (3.8) represents the heat transfer ($\dot{Q}$) that each panel is receiving from the energy sources (the arc and molten surface); it is obtained by implementing the Stefan-Boltzmann law considering the value of the view factors from the arc and molten surface to each panel ($F$) and the emissivity of the surface of the energy sources. The view factors are determined according to ration between each panel's surface area to the total area of the walls ($A_w$) (Table 3-1) and the values of the view factors from arc and molten surface to the walls are shown in Table 3-2. The values of view factors are obtained using the expressions for coaxial parallel disks and outer surface of cylinder to annular disk at end of cylinder geometries (shown in figure 3-5). The radiation emitted from the molten surface to the roof is determined with the
equation (3.9) (Cengel and Ghajar 2014), where $S$ is represented in the equation (3.10), and $A1 = B1 = r_{int}/h_{wall}$, as the view factor limit is the unity, the view factor from molten surface to the walls can be estimated with the equation (3.11). The portion of radiation emitted from the electric arc to the molten surface is calculated with the equation (3.12) (Rea 1975), where $R = (d_{arc}/2)/r_{int}$, $L = h_{arc}/r_{int}$, $A2 = L^2 + R^2 - 1$ and $B2 = L^2 - R^2 + 1$. As the view factor limit is the unity, the radiation from the arc to the rest of surroundings is calculated (3.13); and then the view factor from arc to walls is calculated with the equation (3.14), using a relation of surface areas where the surface area of surroundings is equal to the sum of the roof and walls areas ($A_{surr} = A_{roof} + A_{w}$).

$$
\dot{Q}_{pn} = A_{s1}F_{12pn}\sigma\varepsilon_1(T_{s1}^4 - T_{s2pn}^4) + A_{arc}F_{arcpn}\sigma\varepsilon_{arc}(T_{arc}^4 - T_{s2pn}^4)
$$

(3.8)

Figure 3-5. Geometries used for view factors. A) Coaxial Parallel disks. B) Outer surface cylinder to annular disk.

$$
F_{1-roof} = \frac{1}{2} \left( S - \left( S^2 - 4 \left( \frac{r_{int}}{r_{int}} \right)^2 \right)^{0.5} \right)
$$

(3.9)

$$
S = 1 + \frac{1+B^2}{A1^2}
$$

(3.10)

$$
F_{12} = 1 - F_{1-roof}
$$

(3.11)

$$
F_{arc-1} = \frac{B^2}{8+R+L} + \frac{1}{2\pi} \left( \arccos \left( \frac{A2}{B2} \right) - \frac{1}{2+L} \left( \left( \frac{(A2+2)^2}{R^2} - 4 \right)^{0.5} \right) * \arccos \left( \frac{A2+R}{B2} \right) - \left( \frac{A}{2+R+L} \right) \right)
$$

arcsin(R)

(3.12)
In the equation (3.15), the same portion of heat, from the arc and molten surface, is transferred between the panel layers (slag layer, pipe thickness and water layer), where a global transfer coefficient \( U_{pn} \) for each panel is necessary and is obtained using the equation (3.17). In the equation (3.16) now the heat transferred in each panel is removed by the internal flow represented with an internal energy balance. In these three equations (3.8, 3.15, 3.16), the principal unknowns are the heat transfer \( \dot{Q}_{pn} \), slag surface temperature \( T_{s2} \) and outlet water temperature \( T_{f\text{out}_{pn}} \), which can be solved by a simultaneous equations system.

\[
\dot{Q}_{pn} = U_{pn} A_{pn} \left( T_{s2} - \frac{T_{f\text{in}} + T_{f\text{out}_{pn}}}{2} \right) \quad (3.15)
\]

\[
\dot{m}_{pn} = \dot{m}_{pn} C_{p} (T_{f\text{out}_{pn}} - T_{f\text{in}}) \quad (3.16)
\]

\[
U_{pn} = \frac{1}{A_{pn} \cdot R_{\text{total}_{pn}}} \quad (3.17)
\]

\[
\dot{m}_{pn} = \rho_{\text{water}} \cdot V_{pn} \cdot A_{\text{intpipe}} \quad (3.18)
\]

Flow mass (3.18) is obtained from the flow distribution model. The global transfer coefficient is obtained calculating the total thermal resistance \( R \) (3.19) in the panel layers, which are the resistance by conduction in the slag adhered layer and conduction in thickness of the steel pipes (3.20, 3.21), also another resistance by convection is calculated (3.22) in the water layer inside the pipe, where is necessary to know the heat transfer coefficient by convection inside the panels \( h_{pn} \). Conduction resistances were assumed for a plane wall, because this simplify the analysis and thickness of the steel and slag is small compared with the curvature radius. HTC inside panels is calculated using the Gnielinski's correlation for internal flow (3.23) (Gnielinski 1976), where the friction factor and Re are taken from the
flow-distribution model. The thermal conductive values, lengths, surfaces and rest of parameters used are shown in Table 3-2.

\[ R_{\text{total}pn} = R_{\text{cond slag}pn} + R_{\text{cond steel}pn} + R_{\text{conv fluid}pn} \]  
(3.19)

\[ R_{\text{cond slag}pn} = \frac{L_{\text{slag pn}}}{K_{\text{slag}} A_{pn}} \]  
(3.20)

\[ R_{\text{cond steel}pn} = \frac{L_{\text{steel} pn}}{K_{\text{steel}} A_{pn}} \]  
(3.21)

\[ R_{\text{conv fluid}pn} = \frac{1}{h_{pn} \pi d_{\text{pipe}} L_{pn}} \]  
(3.22)

\[ h_{pn} \cdot d_{\text{pipe}} / K_{\text{water}} = \frac{(f_{pn} / b)^2}{1+12.7 \cdot (f_{pn} / b)^{0.5} \cdot \left( Pr_{\text{water}} \right)^{2/3} \cdot \left( Pr_{\text{water}} - 1 \right)} \]  
(3.23)

To estimate the surface temperature of the pipe (behind the slag) the equation (3.24) is used considering the thermal resistance of the slag layer. The main results that can be obtained are the heat transfer \( (\dot{Q}_{pn}) \), outlet water temperature \( (T_{\text{f out} pn}) \), slag surface temperature \( (T_{s2}) \) and pipe surface temperature for each panel; these are function of the slag-layer thicknesses and arc coverage. The total heat transfer in all EAF walls is calculated with the equation (3.25), a sum of the heat transfer in each panel. Now, the set of equations is complete, and the heat transfer model for each panel can be solved with a simultaneous equations system, same number of equations and unknowns is present, and both models can be solved using the Engineering Equation Solver (EES).

\[ \dot{Q}_{pn} = \frac{T_{s2 pn} - T_{s3 pn}}{R_{\text{cond slag}pn}} \]  
(3.24)

\[ \dot{Q}_{\text{Total}} = \sum_{1}^{14} \dot{Q}_{pn} \]  
(3.25)
Table 3-2. Parameters used in the heat transfer model when using the tubular-panel system.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Wall-Surface Area</td>
<td>56.544 m²</td>
</tr>
<tr>
<td>Surface area of roof</td>
<td>42.74 m²</td>
</tr>
<tr>
<td>Internal radius of EAF</td>
<td>3.68 m</td>
</tr>
<tr>
<td>Walls height</td>
<td>2.44 m</td>
</tr>
<tr>
<td>View Factor from arc to walls</td>
<td>0.3241 (harc = 0.8 m), 0.3047 (harc = 0.4 m), 0.2997 (harc = 0.3 m), 0.2948 (harc = 0.2 m), 0.2898 (harc = 0.1 m)</td>
</tr>
<tr>
<td>View Factor from molten surface to walls</td>
<td>0.478</td>
</tr>
<tr>
<td>Molten surface area</td>
<td>42.74 m²</td>
</tr>
<tr>
<td>Arc diameter</td>
<td>0.15 m</td>
</tr>
<tr>
<td>Stefan-Boltzman constant</td>
<td>5.67E-8 W/m²-K⁴</td>
</tr>
<tr>
<td>Emissivity of the slag</td>
<td>0.8ᵃ</td>
</tr>
<tr>
<td>Emissivity of the arc</td>
<td>0.3ᵇ</td>
</tr>
<tr>
<td>Surface temperature of the arc</td>
<td>10000 Kᵇ</td>
</tr>
<tr>
<td>Inlet Water Temperature</td>
<td>310 K</td>
</tr>
<tr>
<td>Molten surface temperature</td>
<td>1873 K</td>
</tr>
<tr>
<td>Thickness of the pipes</td>
<td>0.01109 m</td>
</tr>
<tr>
<td>Thermal Conductivity of the steel</td>
<td>60.5 W/m-Kᶜ</td>
</tr>
<tr>
<td>Thermal Conductivity of the slag</td>
<td>2.4 W/m-Kᶜ</td>
</tr>
</tbody>
</table>

ᵃ According to (Khodabandeh et al. 2017)
ᵇ According to (Bowman and Krüger 2009)
ᶜ According to (Cengel and Ghajar 2014)

3.3 Results and discussions

3.3.1 Flow distribution in the panels and heat-transfer coefficient

With the two models presented previously, the flow distribution in each panel and the internal HTC in the panels were obtained (figure 3-6) according to the total flow registered. As is shown in figure 3-6, the internal HTC in the panels depends highly to the water flow in piping of each tubular panel. The flow is completely turbulent and
the Reynolds number inside the pipes is around to $5.7 \times 10^5 \sim 8.6 \times 10^5$, as shown in the figure 3-7, also it is shown the panels with higher water velocities, the lower velocities of water in pipes are for the panels 6 and 12 around to 5.5 m/s and the highest for panels 7 and 8 around to 8.1 m/s. Jones et.al. (1998) states that in regions of very high temperatures, the water velocities in the panels piping should be of the order of 5 m/s to ensure that steam bubbles do not form in the pipes. The Panels 7, 8 and 9 are the smallest and the ones with less curvatures (losses), having a high percentage of the total flow (>8.5%), water velocities and larger Reynolds number, showing a higher capacity of heat removal (internal HTC > 37 kW/m²K) over the other panels, the panels 6 and 12 are the ones with more surface area and pipe curvatures (as the configuration of figure 3-3 B), losses, therefore its water flow, Reynolds and the internal HTC are the lowest. A difference of the previous works, it is observed that the water flow and internal HTC varies for each panel, since panels are different, and these values depend on the losses and dimensions of each one.

![Figure 3-6](image.png)

*Figure 3-6. Percentage of total flow in each panel (left column) and internal heat transfer coefficient (right column).*
3.3.2 Effects of slag thickness and arc coverage upon temperatures and heat transfer inside the EAF.

The temperatures of the outer solid slag surface, adhered to the walls, depends highly on the arc coverage and slag layer thickness. The solid slag has a melting temperature around to 1400°C (~1700K) (Guo and Irons 2003), the figure 3-8 shows the temperatures that solid slag can reach according to the arc coverage and slag thickness, it is for the panel 4 which has the similar geometry of the panels 5, 10, 11 and 14, the outer solid slag surface temperatures are very similar to the figure 3-8 for the rest of the panels, the main results obtained in figure 3-8 are the slag-layer thicknesses that can be adhered according to the arc coverage and the slag melting temperature (~1700K). The thickest slag-layer in the process is around to 0.045m.
this happens when the arc is completely covered; and 0.013m of thickness when the arc is having 0.4 m height. As shown in figure 3-8 the outer solid slag temperature increases when the slag is thicker, working as an isolation, the cooling panels remove less heat and therefore the slag adhered to the panels get heated by the radiation sources as shown in figure 3-4.

![Graph showing the relationship between slag layer thickness and outer solid slag surface temperature.](image)

**Figure 3-8.** Surface temperature of slag-layer of panel 4, according to its thickness and arc coverage.

Figure 3-9 shows the outer surface temperatures of the pipes that were obtained for different thicknesses of the slag layer and arc coverages. Outer pipe temperature it can be considered as at the temperature that the steel panels are operating. It is shown in figure 3-9 (A) that by the time the arc height is 0.4 m and there is no slag adhered to the panels, the temperature is around 393 K for the hottest panels that are 6 and 12, and when the arc is completely covered (C) and there is a layer of slag of 0.045m, the maximum temperatures calculated in the surface of pipe are around the 326 K and 325 K for the minimum temperatures.
The surface temperatures of the pipes in the EAF process are not considered to make decisions in the operation of the EAF, as there is no a way of knowing them in the current process. What is considered and based to make decisions in the furnace operations, as to stop the electric current, are the outlet water temperatures in the panels since are monitored in real time and there is a registry of these values, obtained by means of sensors.

In the figure 3-10, the outlet water temperature in each panel was obtained for the different arc coverages and slag layer thicknesses, the thicknesses were taken within the range allowed for the arc coverage used. The thicker the slag layer adhered is, the lower outlet temperature because the slag works as isolation from the radiation of the arc and molten surface. If the height of arc is bigger, the outlet temperatures increase this is because the arc is emitting more radiation. For the case figure 3-10 (A), the difference of temperatures is around to 17 degrees respect to the inlet water temperature (310 K) when there is no slag adhered to the hottest
panels and the difference decrease to 9 degrees when the arc is completely covered; and it goes to 3 degrees if the slag-layer has the maximal thickness (C). The coolest panels have a difference around of 2 degrees against the inlet temperature for the optimal operating conditions of the EAF. As shown in the section 3.3.1, the panels 6 and 12 are the ones with less heat removal capacity (around to 26.6 kW/m²-K) therefore they are the hottest (as shown in figures 3-9 and 3-10). If the water flow is regulated in each panel by control valves, in such a way that the smaller panels have a lower flow, so the larger ones will have a greater capacity to remove the heat and therefore, similar temperatures can be reached in all the panels, avoiding having some of them hotter than others, achieving the uniformity.
Figure 3-10. Comparison of outlet water temperature in the panels with different slag thicknesses. A) Arc height = 0.4 m. B) Arc height = 0.2 m. C) Arc height = 0.0 m.

The heat transfer was also obtained for each panel according to the adhered slag-layer thickness and the height of the arc, seen in figure 3-11. Most panels have the
same surface area, that is why they have very similar values of heat transfer. The panels 1, 7 and 8 are the ones with less surface area, therefore these are receiving less radiation. As seen in figure 3-11 the heat transfer is higher when there are no slag layers adhered to the cooling panels and the arc is not covered, therefore the panels must remove the heat that are receiving directly from the radiation sources. As same as the temperature results, the heat transfer decreases in the panels when the arc is covered in there are thick enough slag-layers adhered. The slag works as an isolation, less heat is passing to the panels and therefore the heat removal is lower.
Figure 3-11. Comparison of the heat transfer in the panels with different slag thicknesses. A) Arc height = 0.4 m. B) Arc height = 0.2 m. C) Arc height = 0.0 m.

In the figure 3-12, it is considered that slag thickness is varying uniformly for all panels to obtain the total heat transfer in the EAF wall, the curves are interrupted
when they reach the maximal thickness limit in the slag-layer according to the arc coverage present. The worst scenario is when the arc is not covered and there is no slag adhered to the walls, the heat transfer can be considered as energy losses by cooling in the EAF process. For the case when height of arc is 0.4 m the losses are over the 17.4 MW just in the wall of the EAF, in the other hand; when the arc is completely covered and the slag layer has its maximal value of thickness (0.045m), the losses are around to the 3.35 MW, considerably reducing losses up to five times of the critical scenario.

![Figure 3-12. Total heat transfer in the cooling panel walls.](image)

3.3.3 Water flow regulation with valves

Trying to improve the uniformity of water temperature in the panels so that all have similar temperatures and their life time/maintenances have similar scheduling in all the panels, a value $K_{L,V_{pm}} = 17$ for loss coefficient is used in the control valves for the panels 7, 8 and 9; which means valves are closed $\frac{3}{4}$ using gate valves. This cannot be considered as control engineering, this is performed only with the purpose
to observe that exists some ways that can be applied to these systems to improve they perform, as an example could be the control engineering in the cooling systems by valves.

Regulating the water flow by closing valves, new water flow distributions and internal HTC in piping of each panel are obtained as shown in the figure 3-13. As the previous results in the figure 3-6 the panels 7, 8 and 9 were the ones with the highest water flow and heat removal capacity. Now by closing the valves or increasing the pressure drop in the panels 7, 8 and 9 (varying the loss coefficient in the valves), the rest of the panels take more percentage of the total flow and therefore its internal HTC increases, the panels 7, 8 and 9 now have a lower percentage of total water flow around to 6% and panels 6, 12 around to 6.65% with an internal HTC around to the 29.5 kW/m²-K; and near to 27 kW/m²-K for panels 7, 8 and 9. Without flow regulation the internal HTC is around to 26.6 kW/m²K for the hottest panels and around to 39 kW/m²-K for the coolest (figure 3-6).

![Figure 3-13. Percentage of total flow in each panel (left column) and Internal Heat Transfer Coefficient (right column) with a $K_{L/VpH} = 17$ for the panels 7, 8 and 9.](image)
Now to see how the uniformity of the temperature in the panels is improved with the flow regulation, the outlet water temperatures in each panel are obtained by closing ¾ the valves of panels 7, 8 and 9. The extreme case in which the furnace may be operating is for an arc height of 0.8 m over the molten surface, the maximal slag-layer thickness for this, is around to 0.007 m, the water temperatures are obtained according to the thickness of the slag adhered to the panels for the cases in which the flow in the panels is not regulated (figure 3-14 A) and when the flow is regulated in the panels 7, 8, 9 (B). When there is no flow control, the outlet water temperature reaches up the 336 K (A) when there is no slag in the panels, that is a temperature difference of 26 degrees respect to the initial temperature of water (310 K). Performing the flow regulation (B) in panels 7, 8, 9 makes all the other panels to improve their conditions to remove heat, and now panels 6 and 12 have a cooler water temperature close to 333 K; also, for the rest of panels there is a slight improvement in their water temperatures, except for the panels where their losses increased.
In addition, the surface temperatures of the pipes become more uniform in all the panels when there exists a flow regulation, just by controlling the valves of panels 7,8,9. In the case (A) figure 3-15 the hottest temperature is when the furnace is in the extreme condition of 0.8m of arc height and there is no slag in the panels, panels 6 and 12 reach temperatures of 436 K without flow regulation; when there is a flow regulation (B) they have temperatures of 434 K, the hottest is panel 9 around to 437 K but the rest of the panels maintain a uniformity in the surface temperature of the pipes, this is because the flow regulation was not accurate and a flow control needs to be applied in all panels.

Remember that an engineering control can be implemented by means of sensors and valves to regulate the flow in each panel to maintain a uniform temperature in all the panels and not just doing it in an arbitrary way like closing ¾ the valves of panel 7, 8 and 9.
The temperatures of the outer surface of the slag-layer adhered to the panels no present big difference when there is a flow regulation present and the proportion of
the slag thickness that can be achieved in the panels for different arc heights is very similar, also the total heat transfer or energy losses in the process are in the same magnitude without presenting a big change, with this can be said that flow control should only be applied if it is wanted that all panels maintain the same temperature, lengthening their life time and not to reduce energy losses.

For the case that the arc has a height of 0.8 m, the total heat transfer or energy losses are around to 26.58 MW when there is no slag adhered to the panels and 20.46 MW for the maximal slag thickness of 0.007 m adhered, it is not at all advisable that the furnace operate under these conditions and for that reason it was considered as a separate case and used to show how there is an improvement in the uniformity of the temperatures when a water flow regulation is performed in the tubular-panel system.

3.3.4 Increase temperatures by decreasing the flow, leaks detection

If water drop is detected in the water flow of a panel piping, it may is due to a water leak and this causes that water temperature starts to increasing; therefore if an increment of temperature is detected can also be said that it is because the water is leaking, and the operators of furnace have instructions to stop the process when there exists very sudden changes in the water temperatures of panels because preventing to the enough water falls into the molten steel, steam explosion can occur. Whit this model the temperatures of the pipes surface and outlet water can be obtained when the water flow in the panels piping is being reduced, to see how it will affect in case of a drop in the water flow in a panel. This is performed for the panel 4 (figure 3-16) which is similar to the panels 5, 10, 11, 14, the water flow is decreased gradually until the temperature conditions still valid for different arc coverages and there is no a slag-layer adhered to the panel leading to the case when the water could impact directly over the molten steel and with the probability to cause an explosion. As it is shown in the figure 3-16, the temperature of pipe surface and water start to increase as the water flow decreases in the panel.
If the operator knows the temperature of the water in each panel, according to this figure 3-16 or decreasing the flow in the model proposed. He can assume if there is a water leak or not, since we can consult directly to the amount of water flow that corresponds to that temperature and deduce how much flow is being lost in the pipes, which is the difference between the initial flow and the current flow (value of the flow according to the current temperature).

As an example, the initial flow that panel 4 should has is around to 22.62 l/s if an operator detects an outlet temperature of 320 K, this panel has an internal flow around to 17.85 l/s when the arc is covered and no slag is adhered to the panel (figure 3-16). There will be a 4.77 l/s of flow drop which could be caused by a leak in the piping and there exists the risk of a steam explosion.

Figure 3-16. Surface and water outlet temperatures in each panel according to the water flow.
3.3.5 Comparing experimental data with model results

The experimental data of the next cases is for information collected when most of the steel has already melted and the arc has a power close to 140 MW.

To verify the results obtained with the models proposed, a comparison of the experimental data against model results is performed (figure 3-17). The experimental values are an average of the outlet water temperatures in each panel for an average arc power of 142 MW, the data was taken for a period of 12 minutes from the operation of the furnace (Data from May-2017); for the panels 1, 6, 12 and 14 there was no experimental data available. The experimental outlet water temperatures are within the range of outlet water temperatures obtained with models when the arc is completely covered (A) and there is a slag-layer adhered to the panels, the standard deviation (SD) is shown for the outlet water temperature in each panel. The standard deviation is the tendency that water temperature in each panel can vary, for the case (A), arc covered, just the panels 8, 10 and 11 need of this parameter to be within the slag thicknesses ranges accepted. Panel 8 is located below the fume duct of EAF and most of the gases inside the furnace pass through this panel and may add more heat to the panel, that is why the tendency of no having slag adhered; in the model, only the radiation from the arc and molten surface are being considered as the heat sources. For the panels 10 and 11, a thicker slag-layer should be present, for the rest of the panels the slag-layer is within the range of 0 cm and 4 cm of thick, so the outlet water temperatures values of the experimental data be within the values of the model when the arc is covered, including the standard deviation of data. The average slag thickness is around to the 20 mm. In the case (B) of figure 3-17, the experimental data is compared now against the results of model when arc is uncovered 0.2 m, the results show that most of water temperatures are within the ranges of model, including the SD, except for the panels 10 and 11 where a slag thicker than 30 mm is needed. The case (A) showed a better relation of the experimental data against the model, so this study can roughly predict what is the
arc coverage and the range of slag thickness that the panels would have. The average energy losses of the studied cases are shown in the table 3-3.

Figure 3-17. Comparison of the experimental data against model results, from May 30, 2017. A) Arc covered. B) Arc uncovered 0.2 m.
An extra comparison was done (figure 3-18) for a different date (October -17), now for average water temperatures of an average power of 137 MW; the data was taken from a period of 16 min of the EAF cycle; for the panels 1, 6 and 12 there was no experimental data available. In the figure 3-18 (A), the experimental data is within the range of outlet water temperatures of panels when arc is covered and for a slag-layer with a thick between 0 and 40 mm. All the experimental outlet water temperature data of each panel match the range of model, considering the standard deviation, being no too good for panels 2 and 8, as the previous data of May-17, the panel 8 is below the fume duct and the tendency to have no adhered slag still present. For the panel 2, there exist the tendency to have a slag-layer thicker than 40 mm. The case (B) shows the comparison when the arc is uncovered 0.2 m, the experimental data also shows a good relation according to the thickness permitted to the arc height (0-30 mm) and it within the range of temperature values, except for the panel 2 where a thicker slag-layer should be present. Both cases showed the different thicknesses of slag and arc coverage that EAF has, which are valid for the collected experimental data, for this, the furnace could have been operating with the arc discovered from 0 to 0.2 m and with a thickness of slag in the walls between 0 and 40 mm. The average slag-layer thickness is around to the 17 mm, the energy losses for the different arc coverages are present in the table 3-3.
Figure 3-18. Comparison of the experimental data against model results, from October 17, 2017. A) Arc covered. B) Arc uncovered 0.2 m.

Another comparison was done (figure 3-19, October 17), now for average temperatures of an average power of 128.88 MW and data taken from a period of 5 min. Unlike the previous comparisons, for the experimental data match the range of
the model results when there are different arc coverages and slag thicknesses, as seen in figure 3-19. For the case (A) the arc should have a height 0.3 m, most of the experimental values are within the range of temperatures according to the slag thickness permitted (0 – 17 mm), except for the panels 2 and 8 where a thicker slag-layer is needed. In the case (B), the arc is uncovered 0.4 m but temperatures are not within the range of slag permitted (0 – 13 mm) for the panels 2, 7, 9 and 10. And case (C) when the arc has a height of 0.2 m over the molten surface, most of the experimental temperatures are within the range of slag permitted (0 – 30 mm), except for the panels 2 and 14, for the panel 2 a thicker slag should be present and for panel 14 which is receiving a greater radiation than an uncovered arc of 0.2 m can emit. Most experimental data match the range of EAF operation for the previous cases presented, so it is deduced that EAF could be operating with different arc coverages, but it is observing that comparisons values have a better correlation for the results when arc height is 0.3 m. The average slag-layer thickness is around to the 12.5 mm, the energy losses for the different arc coverages are presented in the table 3-3.
Figure 3-19. Comparison of the experimental data against model results, from October 17, 2017. A) Arc uncovered 0.3 m. B) Arc uncovered 0.4 m. C) Arc uncovered 0.2 m.

For the previous cases, the average energy losses were measured for the average slag thickness present, shown in table 3-3. The data from May 5, is presenting energy losses around to 4.34%-5.22% of the 142 MW arc power. For the data from
October 17, there are energy losses around to 4.93%-5.90% of the 137 MW. For the last study, also October 17, the losses are around to 7.66% - 9.04% of the 128 MW. Energy losses highly depend on the arc coverage and slag thickness.

Table 3-3. Average energy losses for experimental data with different arc coverages.

<table>
<thead>
<tr>
<th>Date</th>
<th>Avg arc power (MW)</th>
<th>Avg slag thickness (mm)</th>
<th>Arc height (cm)</th>
<th>Heat transfer (MW)</th>
<th>Energy losses</th>
</tr>
</thead>
<tbody>
<tr>
<td>05/30/17</td>
<td>142</td>
<td>20</td>
<td>0</td>
<td>6.16</td>
<td>4.34%</td>
</tr>
<tr>
<td></td>
<td>142</td>
<td>20</td>
<td>20</td>
<td>7.41</td>
<td>5.22%</td>
</tr>
<tr>
<td>10/17/17</td>
<td>137</td>
<td>17</td>
<td>0</td>
<td>6.76</td>
<td>4.93%</td>
</tr>
<tr>
<td>10/17/17</td>
<td>137</td>
<td>17</td>
<td>20</td>
<td>8.27</td>
<td>5.90%</td>
</tr>
<tr>
<td></td>
<td>128.88</td>
<td>12.5</td>
<td>20</td>
<td>9.88</td>
<td>7.66%</td>
</tr>
<tr>
<td></td>
<td>128.88</td>
<td>12.5</td>
<td>30</td>
<td>10.80</td>
<td>8.38%</td>
</tr>
<tr>
<td></td>
<td>128.88</td>
<td>12.5</td>
<td>40</td>
<td>11.65</td>
<td>9.04%</td>
</tr>
</tbody>
</table>

In general, for the cases described above, the slag thickness ranges that the panels must have are shown, so that a certain validity of the proposed model was verified. It was verified that these thicknesses of slag are similar with those indicated by several authors. Khodabandeh et al. (2017) measured the average slag-layer formed on panel with a thickness around to 10 mm. Trejo et al. (2012) affirmed that the ideal thickness of the slag is around to 30 mm, so to slag-layers do not reach their melting point. Also, some slag samples were taken from the EAF wall cooling panels (figure 3-20), with thicknesses of 23.8 mm and 7.1 mm approximately, which are within the range used in this study. Process engineers also confirmed that the slag thickness range, formed in the panels, is not greater than 45 mm for the samples they have taken.

..
Figure 3-20. Slag samples taken from the cooling panels.
Chapter 4. **Spray cooling system**

The models of water flow distribution on spray nozzles piping network and heat transfer of the EAF wall using the spray cooling system are presented. Parameters values are listed in tables in the following sections.

### 4.1 Flow-distribution model

To implement the spray cooling system in the EAF wall, it is necessary to have a piping array with the enough quantity of tangential spray nozzles to cover the total surface area of EAF wall. This system is not yet implemented in the studied EAF, that is why the conditions of the spray configuration are assumed. It is considered the nozzles have a separation distance of 0.1525 m from the wall surface and the form sprayed water is projected is shown in the figure 4-1, using full-cone tangential spray nozzles to carry out the analysis, with a spray angle of 90°. Different types of nozzles that operate with a wide range of angles and pressures can be found with the sellers of these systems from 60°-120° and 10 psi – 100 psi respectively, for this study it was decided to use full cone spray nozzles with a spray angle of 90° for a 20-psi pressure (137895 Pa). The spray cooling systems in the EAFs tend to use pressures around to 20 psi and 30 psi (Ward 1986). Even the sprayed cone leaves a surface without getting directly wet by the tiny droplets, it is considered that the entire wetting surface covered by the spray of each nozzle is the square surface between each nozzle (0.093025 m²), called as nozzle section with a surface area “\( A_{al/bi} \)” for future references, since the array of nozzles are vertical, and water tends to wet the rest of the surface.
To cover the wall surface of the EAF with sprayed water using the above configuration, it is proposed to use 8 pairs of nozzles (a and b for each pair) in a piping network array, which is fed from the top with an initial pressure ($P_o$). As shown in figure 4-2, the piping network is consisted of a main pipe with a diameter ($d_o$) and it is connected to secondary pipes, which distribute the water flow to the nozzles. Secondary pipes have a smaller diameter ($d_i$) and a length ($L_i$). Each nozzle has a separation between them $L_o$ and a height ($z_i$), this by considering the impact area of the spray. With this piping network configuration, the height of the wall is covered (2.44 m) with a width of 0.61 m, therefore 38 piping network array configurations, as the figure 4-2, are necessary to cover the entire EAF wall surface with the sprayed water. Hence, 608 spray nozzles are needed to cover the entire EAF wall.
To clarify the configuration of nozzles covering the wall surface of the EAF, spray cooling system, the figure 4-3 shows the arrays of pipes around the EAF wall, discharging the water flow through the spray nozzles and this impacting directly to the surface of the wall, which is a steel plate like the system implemented in the figure 1-5. With this model is pretended to find the water flow that is crossing through each nozzle.
The assumptions of this model are as follows:

- Steady state
- Water properties constant and taken at a 310-K temperature.
- All the piping network configurations have an initial pressure of 137.895 kPa (20 PSI), pressure losses in the flow path to feed each array are neglected.
- The flow is discharged to an atmospheric pressure.

From an individual energy balance on each nozzle, the head losses can be represented with the equation (4.1), for the section of pipe that connects to each nozzle of the sets “a” and “b” \( (H_{La_i}, H_{Lb_i}) \), which depends on the initial pressure \( (P_o) \), velocity of water \( (V_{ai}, V_{bi}) \) and the height in which each nozzle is located \( (z_i) \), a correction factor \( (\alpha) \) for turbulent flow is used with a value of 1.05.
Chapter 4

\[ H_{Lai/Lbi} = \frac{P_o}{\rho_{\text{water}} g} - \left( \alpha * \frac{v_{ai/bi}^2}{2g} + z_i - z_o \right) \]  \hspace{1cm} (4.1)

The distribution of the water flow in the piping network is set in the equation (4.2), as shown in the figure 4-2 the flow is being distributed to sections “ai” and “bi” (a1,a2,..,a8 and b1,b2,..,b8) after it goes through the sections of the main tube (o1,o2,..,o8).

\[ \dot{V}_{oi} = \dot{V}_{ai} + \dot{V}_{bi} + \dot{V}_{oj} \]  \hspace{1cm} (4.2)

Where:

\[ j = i + 1 \]
\[ \dot{V}_{oj} = 0, \quad j > 8 \]

The head losses can also be estimated by the major losses in addition with minor losses along each section of the pipes, the equation (4.3) is used to obtain the head losses of each pipe sections that connects the spray nozzles. Major losses depend mainly on the friction factor of each section and wet length, while minor losses depend on the loss coefficients due to junctions \( (K_{TL}) \) and nozzles. The loss coefficient for each spray nozzle was obtained with the work of Alvi et al. (1978), in this work the losses coefficients of nozzles are determined according to the ratio between nozzle outlet diameter and inlet diameter \( (\beta) \) for Reynolds numbers up to 10,000; where the flow is completely turbulent for internal flow conditions. They assume that the loss coefficient is substantially constant for Reynolds numbers greater than 10,000. The ratio of diameter of the nozzle proposed is 0.46 and the loss coefficient of the spray nozzle \( (K_{\text{nozzle}}) \), extrapolated from Alvi’s work, is 17.835 for a Reynolds number greater than 10,000 (turbulent), adding the losses of the branch flow the loss coefficient of spray nozzles is equal to 18.835.
\[ H_{Lai/Lbi} = \left( \frac{f_{o1} \cdot L_{o1}}{d_o} \right) \cdot \left( \frac{V_{o1}^2}{2 \cdot g} \right) + \sum_{j=1}^{i} \left( \frac{f_{oj} \cdot L_{oj}}{d_o} + K_{TL} \right) \cdot \left( \frac{V_{oj}^2}{2 \cdot g} \right) + \left( \frac{f_{ai/bi} \cdot L_i}{d_i} + K_{nozzle} \right) \] 

\[ \left( \frac{V_{ai/bi}^2}{2 \cdot g} \right) \] 

(4.3)

Where:

\[ V_{oj} = 0, \quad j = 1 \]

The main unknown variable in the above equation are the friction factors \( f \) of each section, which are obtained with the Colebrook’s equation (Colebrook et al. 1939). The friction factors for the sections of the main pipe \( (o1,o2,…,o8) \) are obtained with the equation (4.4) where their Reynolds numbers are needed (4.5). For the sections that connect to nozzles, the friction factor is obtained with the equation (4.6), which needs the Reynolds number from (4.7) and material roughness \( e \).

\[ \frac{1}{f_{oi}^{0.5}} = -2 \cdot \log 10 \left( \left( \frac{2.51}{Re_{oi} \cdot f_{oi}^{0.5}} \right) + \left( \frac{e}{d_o \cdot 3.7} \right) \right) \] 

(4.4)

\[ Re_{oi} = \frac{\rho_{water} \cdot V_{oi} \cdot d_o}{\mu_{water}} \] 

(4.5)

\[ \frac{1}{f_{ai/bi}^{0.5}} = -2 \cdot \log 10 \left( \left( \frac{2.51}{Re_{ai/bi} \cdot f_{ai/bi}^{0.5}} \right) + \left( \frac{e}{d_i \cdot 3.7} \right) \right) \] 

(4.6)

\[ Re_{ai/bi} = \frac{\rho_{water} \cdot V_{ai} \cdot d_i}{\mu_{water}} \] 

(4.7)

Finally, to complete the set of equations for the flow distribution model of the spray cooling system, the mean velocity of each section is determined with the equations (4.8,4.9). The total volumetric flow for a single array of pipes (8 pairs of nozzles) is calculated with the equation (4.10), summation of the flow that passes through each nozzle. Same number of unknows and equations are obtained, the model can be solved with an equation solver. Results for head losses, velocities, and volumetric flows in each section can be found.
\[ V_{o'i} = \frac{V_{o'i}}{\pi\frac{d_i^2}{4}} \]  
(4.8)

\[ V_{ai/bi} = \frac{V_{ai/bi}}{\pi\frac{d_i^2}{4}} \]  
(4.9)

\[ \dot{V}_{\text{total}} = \sum_{i=1}^{8} \dot{V}_{ai} + \sum_{i=1}^{8} \dot{V}_{bi} \]  
(4.10)

**Table 4-1. Parameters used in the flow-distribution model and dimensions of the nozzle sections.**

<table>
<thead>
<tr>
<th>Nozzles/Sections</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L_i ) (m)</td>
<td>0.1525</td>
<td>0.1525</td>
<td>0.1525</td>
<td>0.1525</td>
<td>0.1525</td>
<td>0.1525</td>
<td>0.1525</td>
<td>0.1525</td>
</tr>
<tr>
<td>( L_{ai} ) (m)</td>
<td>0.1525</td>
<td>0.305</td>
<td>0.305</td>
<td>0.305</td>
<td>0.305</td>
<td>0.305</td>
<td>0.305</td>
<td>0.305</td>
</tr>
<tr>
<td>( z_i ) (m)</td>
<td>2.135</td>
<td>1.83</td>
<td>1.525</td>
<td>1.22</td>
<td>0.915</td>
<td>0.61</td>
<td>0.305</td>
<td>0</td>
</tr>
<tr>
<td>( d_i ) (m)</td>
<td>0.0157</td>
<td>0.0157</td>
<td>0.0157</td>
<td>0.0157</td>
<td>0.0157</td>
<td>0.0157</td>
<td>0.0157</td>
<td>0.0157</td>
</tr>
<tr>
<td>( A_{ai/bi} ) (m²)</td>
<td>0.093</td>
<td>0.093</td>
<td>0.093</td>
<td>0.093</td>
<td>0.093</td>
<td>0.093</td>
<td>0.093</td>
<td>0.093</td>
</tr>
</tbody>
</table>

\( K_{\text{nozzle}} = 18.835 \quad K_{T_L} = 0.2 \quad z_o = 2.44 \text{ m} \quad P_o = 137895 \text{ Pa} \quad d_o = 0.0266 \text{ m} \quad e = 0.045\times10^{-3} \text{ m} \quad \alpha = 1.05 \)

**4.2 Heat transfer model**

A heat-transfer model is established for the EAF walls when using the spray cooling system. The main assumption are as follows:

- Steady state.
- The EAF is cylindrical.
- The heat transfer from the molten surface and electric arc to the walls is only given by the radiation mechanism.
- The heat received from the arc and molten surface is completely removed by convection of the spray cooling system.
- The radiation from the electrodes and the convection heat of the gases are neglected.
- All radiation transferred from the arc and molten surface is absorbed by the walls, not reflected.
- The molten surface has a temperature of 1,873 K (1,600 °C) and the electric arc has a temperature of 10,000 K.
- Multiple nozzles arrays effect is neglected, convection coefficient is estimated for each single nozzle.
• Liu et al. (2017) proved that spray angle for full cone spray tends to get narrower when the injection pressure is increased. For this study as the initial pressure is not getting increased, it is considered that the spray angle is not affected by the pressure drops in the piping network and remains constant.
• Single phase regime is considered for the conditions of the spray cooling.

The figure 4-4 shows the elements used to model heat transfer in the EAF-walls when using the spray cooling system. The wall of the EAF is a steel plate, with a thickness ($L_{plate}$) which is cooled by the spray system from the outside of EAF, as seen in figure 4-4. In this model, the equations are established to find the heat transfer and temperatures for each nozzle sections according to the sprayed flow that impacts on them.

![Figure 4-4. Elements of the EAF used in the heat-transfer model for the spray cooling system.](image-url)
The model is represented by the three principal equations (4.11, 4.12 and 4.13). Equation (4.11) represents the heat transfer \( \dot{Q} \) to each nozzle section from the arc and molten surface radiation emitted; it is obtained by implementing the Stefan-Boltzman law considering the view factor from the arc and molten surface to each nozzle section (F) and the emissivity of the surface of the energy sources. The view factors are determined according to the ratio between each nozzle surface section to the total area of wall (A) (Table 4-1) and the values of the view factors from the arc and molten surface to the walls are the same as those obtained for the tubular-panel system analysis (Table 4-2). In the equation (4.12), the same portion of heat, from the arc and molten surface, is transferred between the wall layers of each nozzle section (slag layer, steel plate thickness), where a global transfer coefficient \( U \) for each nozzle section is necessary and is obtained using the equation (4.14). In the equation (4.13), the heat transferred to each nozzle section is removed by the external forced convection of the sprayed water and the heat transfer coefficient in each nozzle section must be calculated. In these three equations (4.11, 4.12 and 4.13), the principal unknowns are the heat transfer \( \dot{Q}_{ai}, \dot{Q}_{bi} \), slag-surface temperature \( T_{s2ai}, T_{s2bi} \), and the outer plate surface temperature \( T_{s4ai}, T_{s4bi} \).

\[
\begin{align*}
\dot{Q}_{ai/bi} &= A_{s1} F_{12 ai/bi} \sigma \varepsilon_1 \left( T_{s1}^4 - T_{s2ai/bi}^4 \right) + A_{sarc} F_{arc ai/bi} \sigma \varepsilon_{arc} \left( T_{arc}^4 - T_{s2ai/bi}^4 \right) \\
\dot{Q}_{ai/bi} &= U_{ai/bi} A_{ai/bi} \left( T_{s2 ai/bi} - T_{s4 ai/bi} \right) \\
\dot{Q}_{ai/bi} &= h_{ai/bi} A_{ai/bi} \left( T_{s4 ai/bi} - T_{fin} \right) \\
U_{ai/bi} &= \frac{1}{A_{ai/bi} + R_{total ai/bi}}
\end{align*}
\]

(4.11) (4.12) (4.13) (4.14)

The global transfer coefficient is obtained by calculating the total thermal resistance \( R \) (4.15) in the layer of wall; this comprises the conduction resistances in the slag-adhesion layer and the steel plate (4.16, 4.17). The conduction resistances were assumed for a plane wall, to simplify the analysis, and the thicknesses of the steel plate and slag layer are small compared with the curvature radius of the EAF.
The external heat transfer coefficient for each nozzle section \( (h_{ai/bi}) \) is determined according to the mean droplet diameter and mean volumetric flux \( (\bar{V}_{ai/bi}) \), by using the correlation for single-phase of Mudawar and Valentine (M&V) (1989) in the equation (4.18), for full cone spray nozzles with vertically downward oriented spray.

As shown in section 2.1.3.1, the effect of spray orientation, either vertical or horizontal, has a minimal influence. In some studies, as the one of Hunnell et al. (2006), was found that horizontal oriented spray can be a slightly better than vertical oriented for cooling. Hence, it was decided to work with the proposed correlations of vertical oriented spray even when horizontal spray is impacting in the EAF walls.

The Nusselt and Reynolds numbers of each nozzle surface section are in function of the Sauter mean diameter of the droplets that are impacting each nozzle section \( (d_{32_{ai/bi}}) \) and can be obtained with the equations (4.19 and 4.20), the mean volumetric flux and Sauter mean diameter are used as the velocity and characteristic length. Properties of sprayed water \( (\rho_{water}, Pr_{water}, \mu_{water}, K_{water}) \) are obtained for a bulk temperature of 315 K. The correlation (4.18) is valid for a mean volumetric flux around to \( (0.6 \times 10^{-3} -10^{2}) \) m\(^3\)s\(^{-1}\)/m\(^2\) and a Prandtl number between 2.06-6.33.

\[
Nusselt_{ai/bi} = 2.512 \times (Re_{Sal/sbi}^{0.76}) \times (Pr_{water}^{0.56}) \tag{4.18}
\]

\[
Nusselt_{ai/bi} = \frac{h_{ai/bi} \times d_{32_{ai/bi}}}{K_{water}} \tag{4.19}
\]

\[
Re_{Sal/sbi} = \frac{\rho_{water} \times \bar{V}_{ai/bi} \times d_{32_{ai/bi}}}{\mu_{water}} \tag{4.20}
\]
The droplet mean diameter, is one of the main factors that affects heat transfer in spray cooling systems. The correlation of Mudawar and Valentine to obtain heat transfer coefficient of the spray for single-phase (4.18), uses the Sauter mean diameter, which also can be determined with a correlation proposed by Estes and Mudawar (1995) for full-cone sprays according to the Weber and Reynolds numbers based on the nozzle’s orifice diameter \(d_{or}\), set in the equations (4.21, 4.22 and 4.23). The properties were considered for a temperature of 310 K. The Weber and Reynolds numbers based on orifice diameter, use the pressure drop of each nozzle which is obtained with the equation (4.24), in function of the head losses from flow distribution model.

\[
\frac{d_{32ai/bi}}{d_{or}} = 3.67 \times \left( W_{e_{oai/obi}}^{0.5} \times R_{e_{oai/obi}} \right)^{-0.259} \tag{4.21}
\]

\[
W_{e_{oai/obi}} = \frac{\rho_{gas} \left( \frac{2 \Delta P_{ai/bi}}{\rho_{water}} \right) \cdot d_{or}}{\sigma_{water}} \tag{4.22}
\]

\[
R_{e_{oai/obi}} = \frac{\rho_{water} \left( \frac{2 \Delta P_{ai/bi}}{\rho_{water}} \right)^{0.5} \cdot (d_{or})}{\mu_{water}} \tag{4.23}
\]

\[
\Delta P_{ai/bi} = H_{Lai/Lbi} \times \rho_{water} \times g \tag{4.24}
\]

The mean volumetric flux over each nozzle section, is obtained with the total volume flow rate in each spray nozzle (taken from flow distribution model) divided by the portion of the impacted surface area, represented in the equation (4.25) (Liang and Mudawar 2017).

\[
\bar{V}_{ai/bi} = \left( \frac{\dot{V}_{ai/bi}}{A_{ai/bi}} \right) \tag{4.25}
\]

To estimate the surface temperature of the plate behind the slag, in each nozzle section, equation (4.26) is used considering the resistance of the slag layer. The final temperature of the sprayed water which is impacting the steel plate surface, can be determined for the entire sprayed surface wall \(T_{f_{outsw}}\) by a balance of energy with the equation (4.27), where the total mass flow of the spray system is obtained with
the equation (4.28) from the volumetric flow passing through each nozzles array. The total heat transfer in all the EAF wall is obtained using the equation (4.29), a sum of the heat transfer in each nozzle section multiplying by the number of arrays of nozzles necessary to cover the entire EAF wall. The set of equations is complete, and heat transfer model can be solved with simultaneous system of equations; same number of equations and unknowns are present, both models can be solved with the Engineering Equation Solver.

\[
\dot{Q}_{ai/bi} = \frac{T_{s2_{ai/bi}} - T_{s3_{ai/bi}}}{R_{stagai/bi}} \quad (4.26)
\]

\[
\dot{Q}_{\text{totals}} = \dot{m}_{\text{totals}} C_p (T_{foutsw} - T_{fin}) \quad (4.27)
\]

\[
\dot{m}_{\text{totals}} = \rho_{\text{water}} \cdot \dot{V}_{\text{total}} \cdot 38 \quad (4.28)
\]

\[
\dot{Q}_{\text{Totals}} = 38 \cdot (\sum_i^8 \dot{Q}_{ai} + \sum_i^8 \dot{Q}_{bi}) \quad (4.29)
\]

An alternate correlation can be used to estimate the external heat transfer coefficient for the single-phase regime in the spray cooling, Rybicki and Mudawar (R&M) (2006) proposed the correlation of the equation (4.30) combining an upward-facing and downward-facing spray also in function of the Sauter mean droplet diameter.

\[
Nu_{stag}R_{ai/bi} = 4.7 \cdot (Re_{stag/bi}^{0.61}) \cdot (Pr_{\text{water}}^{0.32}) \quad (4.30)
\]

As discussed previously in section 4.1, it is considered that the area covered by the spray is the square section between each nozzle, without leaving uncovered spaces by sprayed water in the model. The uncovered surface area of the square section (blank space) is around to the 21.46% of the square surface as shown in figure 4-5 (A). To validate the above assumption, blank space was assumed with an external coefficient of 0 W/m²-K, then an average HTC was obtained by adding the external HTC of the circular area. The new HTC has a 6% lower than the HTC calculated with the entire square surface (figure 4-5 B). If the blank space is assumed with an external HTC of 5000 W/m²-K, the average HTC of the square surface has a
difference of 1% minor than the one using the entire square surface as the sprayed surface, therefore the total coverage of square surface assumption is valid.

![Figure 4-5. A) Surface area covered by spray. B) Sprayed surface assumed.](image)

Table 4-2. Parameters used in the heat-transfer model for the spray system.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Wall-Surface Area</td>
<td>56.544 m²</td>
</tr>
<tr>
<td>View Factor from arc to walls</td>
<td>0.3047 (harc = 0.4 m), 0.2997 (harc = 0.3 m), 0.2948 (harc = 0.2 m), 0.2898 (harc = 0.1 m)</td>
</tr>
<tr>
<td>View Factor from molten surface to walls</td>
<td>0.478</td>
</tr>
<tr>
<td>Molten surface area</td>
<td>42.74 m²</td>
</tr>
<tr>
<td>Arc diameter</td>
<td>0.15 m</td>
</tr>
<tr>
<td>Stefan-Boltzman constant</td>
<td>5.67E-8 W/m²·K⁴</td>
</tr>
<tr>
<td>Emissivity of the slag</td>
<td>0.8ᵃ</td>
</tr>
<tr>
<td>Emissivity of the arc</td>
<td>0.3ᵇ</td>
</tr>
<tr>
<td>Surface temperature of the arc</td>
<td>10000 Kᵇ</td>
</tr>
<tr>
<td>Inlet Water Temperature</td>
<td>310 K</td>
</tr>
<tr>
<td>Molten surface temperature</td>
<td>1873 K</td>
</tr>
<tr>
<td>Thickness of the steel plate</td>
<td>0.0127 m</td>
</tr>
<tr>
<td>Thermal Conductivity of the steel</td>
<td>60.5 W/m·Kᶜ</td>
</tr>
<tr>
<td>Thermal Conductivity of the slag</td>
<td>2.4 W/m·K</td>
</tr>
<tr>
<td>Nozzle’s orifice diameter</td>
<td>7.267E-3 m</td>
</tr>
</tbody>
</table>

ᵃ According to (Khodabandeh et al. 2017)
ᵇ According to (Bowman and Krüger 2009)
ᶜ According to (Cengel and Ghajar 2014)
4.3 Results and discussions

For the nozzle configuration array proposed in figures 4-2 and 4-3 around the entire wall of the EAF, it can be seen how the wall is divided into 8 zones (shown in figure 4-5) according to the pair of nozzles distribution. Under steady operational conditions each zone maintains a uniform capacity for heat removal throughout the EAF. The next results may be described by the zone number in which the EAF wall is divided when using the spray cooling system.

![Sprayed water zones in the EAF wall.](image)

4.3.1 Validation of the heat transfer coefficient correlations

From the models of flow distribution and heat transfer in the spray cooling system, it is proven that the mean volumetric flux is within the range to validates the
correlations used to obtain the external heat transfer coefficients, shown in table 4-3, Prandtl number of the sprayed water is around to 4.7, matching the range of validation. Also, the Sauter mean diameter of the droplets impacting each nozzle section is obtained, the sprayed droplets diameter has a big influence in the heat transfer removal capacity, the smaller the drop size, the greater its capacity to remove heat. For this case there is no a big difference in the droplets diameters between each nozzle. The principal factor that makes the difference of temperatures into the zones of the EAF walls, is the mean volumetric flux over the surface of each nozzle section.

<table>
<thead>
<tr>
<th>Nozzles ai/bi</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>(d_{32}) (10^{-3} m)</td>
<td>1.147</td>
<td>1.137</td>
<td>1.129</td>
<td>1.122</td>
<td>1.115</td>
<td>1.109</td>
<td>1.103</td>
<td>1.098</td>
</tr>
<tr>
<td>(\dot{V}_{ai/bi}) (10^{-3} m^3 s^{-1}/m^2)</td>
<td>7.457</td>
<td>6.452</td>
<td>5.703</td>
<td>5.193</td>
<td>4.897</td>
<td>4.783</td>
<td>4.806</td>
<td>4.912</td>
</tr>
</tbody>
</table>

### 4.3.2 Flow distribution in each nozzle and heat transfer coefficients

With the two models presented previously, the water flow distribution in the nozzles array proposed and its external heat transfer coefficient in the nozzles sections were obtained. The total volumetric flow in each array of nozzles obtained, after solving the flow distribution model, was of 0.008224 m^3/s (130.367 GPM), the figure 4-6 shows the relation of the water flow in each nozzle according to the total flow in the nozzles array proposed. A water flow of 10.99 GPM (8.435% of total flow) is passing through each nozzle of the first pair, then it starts decreasing for the next pairs of nozzles until a certain stability is observed from the fifth pair (7.22 GPM for each nozzle) with a small increase of flow for the last pair. It was observed that the water flow is not uniform throughout the nozzle array, therefore the EAF wall temperature will not be uniform either.

As mentioned previously, in section 4.3.1, the external HTC in each nozzle section is related to the volumetric flux (volumetric flow/surface area). The HTC was
obtained for the different correlations presented for the external convection of the spray, there is no a big difference in the values obtained for both correlations. As shown in figure 4-6, the nozzle section of the first pair of nozzles, zone 1, has the highest value of the external HTC, around to 22.027 kW/m²·K for the M&W correlation and 19.723 kW/m²·K for the R&M correlation since the first nozzle have the highest percentage of flow. The external HTC decreases as the flow in the nozzles does, getting to certain stability upon the fifth pair of nozzles or zone 5 with a HTC around to 16 kW/m²·K and 15.5 kW/m²·K for M&V and R&M correlations, respectively. Nevertheless, the HTC values using the correlation of Mudawar and Valentine (1989) are better related to the changes of flow in each nozzle, therefore it is the one selected to present the next results, in general, the one used to solve the heat-transfer model.

Figure 4-7. Percentage of flow in each nozzle according to the total flow in the nozzles array (left column) and its external heat-transfer coefficient according to the correlations used (right column).
4.3.3 Effects of slag thickness and arc coverage upon surface temperatures, heat transfer and final water temperatures in the EAF wall.

The temperature of the outer solid-slag surface, adhered to the steel plate wall, depends highly upon the arc coverage and slag thickness. The slag has a melting point around to the 1400°C (Guo and Irons 2003). The figure 4-7 shows the temperatures of the solid slag against its thickness according to the arc coverage. The results are taken for the “zone 6” of the wall, which is the one with the poorest heat removal capacity. Nevertheless, the temperature for the solid slag, in the other zones, is very similar and therefore same slag thicknesses can be obtained in the entire wall. The thickest slag layer is around to the 0.045 m, when is arc is covered. The slag layer thickness is around to 0.013 m, when the arc has a height of 0.4 m. As shown in the figure 4-7, the temperature of the slag starts to increase when it gets thicker, working as an insulator, causing less heat to pass into the walls.

![Figure 4-8. Surface temperature of the slag layer in the section of the nozzles “a6” and “b6”, zone 6, according to its thickness and arc coverage.](image)
In figure 4-8, the inner surface temperatures of the plate are obtained, surface behind the slag layer, according to the slag-layer thickness and the height of arc. The figure 4-8 shows the case when the arc is not covered with a height of 0.4 m over the molten surface (A), the case (B) when the arc’s height is 0.2 m and (C) when the arc is completely covered. The maximum temperature reached in the wall, is when the arc is not covered and there is no a slag layer adhered to the wall, the hottest wall section is the “zone 6” with a temperature around to 412.3 K, presenting a considerable uniformity in the zones 5,7 and 8 since the external HTC and water flow rate values are very close. For the case (C) when the arc is completely covered, and the slag layer adhered has the maximal thickness of 0.045 m, the inner temperature decreases considerably around to 329.8 K for the hottest zones (82.5 degrees of difference respect to critical scenario). The “zone 1” is the one with the lowest temperatures in the different cases, because it is the zone with the highest value of flow rate and external HTC, presenting a difference of 6.6 degrees respect to hottest zones when the arc’s height is 0.4 m and there is no a slag layer adhered, and a difference of 1.3 degrees to the optimal scenario. It is observed how improving the operating conditions of the furnace, the temperature in the wall is more uniform.
Figure 4-9. Comparison of the plate inner surface temperature in the nozzle sections for different slag thicknesses. A) Arc height = 0.4 m. B) Arc height = 0.2 m. C) Arc height = 0.0 m.

In figure 4-9, the temperatures of the outer plate surface are obtained, surface in direct contact with the spray. Unlike the inner surface temperatures, the outer surface of the plate present bigger changes of temperature according to section of
the wall. As previous results, the “zone 5” still being the section with the highest outer temperatures and “zone 1” with the lowest temperatures. In the case (A), the highest temperature is around to 333.8 K when the arc has a height of 0.4 m and no slag is adhered in the zone 6, the zone 1 is having a temperature around to 327.1 K for the same conditions, a difference of 6.7 degrees. For the case (C) when the arc is covered, and the maximal thickness of slag is present, the temperature of the hottest zone is reduced to 314.7 K and the coldest zone to 313.4 K. It can be seen, under optimal operation conditions the outer surfaces temperatures are considerably small, which pose no danger to working people nearby to the furnace.
Figure 4-10. Comparison of the plate outer surface temperature in the nozzle sections for different slag thicknesses. A) Arc height = 0.4 m. B) Arc height = 0.2 m. C) Arc height = 0.0 m.

The heat transfer, energy losses of the process, was also obtained for each nozzle section according to the slag-layer thickness and height of the arc. The heat that is being removed by each nozzle is shown in figure 4-10. As all the sprayed surfaces
by the nozzles has the same area, the heat transfer obtained are very similar, negligible difference, so can be said there is a uniformity heat removal around the entire EAF wall even when the flow and external HTC in each nozzle section are different, these do not make a big difference to the amount of heat that is being removed by each nozzle. The heat transfer in each nozzle is greater when no slag adheres to the plate wall and the arc is not covered, meaning the spray cooling system must removes all the heat coming from the radiation sources. For the case (A) arc height of 0.4 m when no slag is adhered, the heat transfer is around to 34794 W per nozzle, in the case (B) for no slag adhered the heat transfer is around to 26507 W per nozzle and in the case (C) also for no slag adhered, the heat transfer is around to 18740 W; and 6693 W per nozzle if the slag layer has a thickness of 0.045 m, the losses are reduced more than 5 times to the critical scenario of the first case (A). The slag-layer benefits the efficiency of the process avoiding that cooling system to remove the heat radiation.
Figure 4-11. Comparison of the heat transfer in each nozzle section for different slag thicknesses. 
A) Arc height = 0.4 m. B) Arc height = 0.2 m. C) Arc height = 0.0 m.

In the figure 4-11, the total heat transfer removed by the entire spray cooling system, the 38 arrays (608 nozzles), is obtained assuming the slag thickness is varying
uniform in all the plate wall for different arc coverages. The curves are interrupted when they reach the maximal thickness limit of the slag for a given arc height. The critical scenario is when the arc is uncovered 0.4 m and no slag is present, the spray cooling removes all the amount of radiation heat reflected to the wall. The heat transfer can be considered as the energy losses of the process; for the critical scenarios the losses by spray cooling are over 21.1 MW; for the optimal case, when arc is covered, and the slag adhered has the maximally thick, losses are around to 4.06 MW, reducing the losses by up to five times.

![Figure 4-12. Total heat transfer in the sprayed cooled wall.](image)

The final temperature of the sprayed water is obtained in the figure 4-12, assuming same conditions of the total heat transfer results, since the sprayed water at the end is a mixed with the water of each zone, so it is a better approximation determining the final temperature according to the total heat removed in the wall. The maximum temperature of the sprayed water is around to 326.4 K when the arc height is 0.4 m and no slag is adhered; a difference of 16.4 degrees respect to initial water
temperature (310 K); the temperature decreases to 320.7 K if the slag layer thickness is 0.013 m. For the case when arc is covered, the temperature of water reaches the 318.9 K when no slag is adhered and decreases to the 313.3 K when the slag is having the maximal thick, around to 3.3 degrees of difference respect to initial temperature.

![Figure 4-13. Final temperature of the sprayed water around to the entire EAF wall.](image)

**4.4 Jet impingement and droplets impingement**

Jet nozzles are used instead of spray nozzles to see the cooling capacity of impinging jets. The water flow rate obtained in the first pair of nozzles is the one assumed for the jet nozzles. The water properties are at 310 K.

The heat transfer coefficient of jet is obtained with the equation of Schlünder and Gnielinski (1967), they correlated single round nozzles for impinging jet in single-phase with the equation (4.31), in function of nozzle diameter, separation \( H \) and
radius of impacted area \((r_{sj})\). \(G_{jet}\) and \(F_{jet}\) are functions obtained with the equations (4.32 and 4.33). The heat transfer coefficient is obtained with the equation (4.34).

\[
\frac{Nu_{jet}}{Pr_{water}^{0.42}} = G_{jet} \times F_{jet} \tag{4.31}
\]

\[
G_{jet} = \left(\frac{d_{or}}{r_{sj}}\right) \times \left(\frac{1-1.1 \times \left(\frac{H}{d_{or}}\right) - \left(\frac{d_{or}}{r_{sj}}\right)}{2.5 \times \left(\frac{d_{or}}{r_{sj}}\right)}\right) \tag{4.32}
\]

\[
F_{jet} = 2 \times \left(Re_{jet} \times \left(1 + \left(\frac{Re_{jet}^{0.55}}{200}\right)^{1.6}\right)\right) \tag{4.33}
\]

\[
h_{jet} = Nu_{jet} \times \frac{K_{water}}{d_{or}} \tag{4.34}
\]

The Reynolds number of the jet is obtained with the equation (4.36) where is necessary to obtain the velocity of jet, which is obtained with the equation (4.35).

\[
V_{jet} = V_{a1} \times \left(\frac{d_{or}^2}{d_{or,1}^2}\right) \tag{4.35}
\]

\[
Re_{jet} = \rho_{water} \times V_{jet} \times \frac{d_{or}}{\mu_{water}} \tag{4.36}
\]

The correlation is valid for \(2 \times 10^3 < Re < 4 \times 10^5\), \(2.5 < r_{sj}/d_{or} < 7.5\) and \(2 < H/d_{or} < 12\). The values of the parameters used, and results of the HTC are listed in the table 4-4.

<table>
<thead>
<tr>
<th></th>
<th>H (m)</th>
<th>Radius (m)</th>
<th>Re</th>
<th>(r_{sj}/d_{or})</th>
<th>(H/d_{or})</th>
<th>HTC (W/m²-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0.1525</td>
<td>0.1525</td>
<td>174486</td>
<td>20.98</td>
<td>20.98</td>
<td>12518</td>
</tr>
<tr>
<td>B</td>
<td>0.05</td>
<td>0.05</td>
<td>174486</td>
<td>6.88</td>
<td>6.88</td>
<td>35810</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>18762</td>
<td>6.88</td>
<td>6.88</td>
<td>7789</td>
</tr>
<tr>
<td>C</td>
<td>0.08</td>
<td>0.054</td>
<td>174486</td>
<td>7.43</td>
<td>11.01</td>
<td>31904</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>48068</td>
<td>7.43</td>
<td>11.01</td>
<td>7669</td>
</tr>
</tbody>
</table>
As shown in table 4-4 for the case (A), similar conditions of the spray nozzle, flow and covered area, the HTC is around to 12518 W/m²-K, anyway this value cannot be taken as true because it does not comply with the validation. The first pair of spray nozzles, or zone 1, has a HTC of 22027 W/m²-K. In the case (B), it is proposed the jet nozzle is separated 0.05 m and covering a surface of the same radius, given a HTC of 35810 W/m²-K when the jet nozzle has the same flow of the spray nozzle; the correlation is valid, however there is no a big difference in the temperature results if nozzles with that HTC are used in the spray system, but 9.3 jet nozzles are required for replacing the surface area that one spray nozzle covers. For the second condition of case (B) the flow is now reduced to 9.3 times as 9.3 nozzles will be required, the HTC is 7789 W/m²-K, it is considerably lower than using spray nozzles, about 2.8 times. In the case (C), 7.97 nozzles are required to cover the zone of the spray nozzle, the HTC is around to 31904 W/m²-K similar to first case. Reducing the flow 7.97 for the configuration (C), the HTC for jet is around to 7669 W/m²-K, 2.87 times lower than the one of the first spray nozzles.

To perform a fairer comparative analysis, similar operating conditions are assumed for jet and spray cooling systems. The heat transfer coefficient was obtained in relation to the area covered by either type of nozzles, jet or spray. Figure 4-14 shows the results obtained, showing the ratio of the heat transfer coefficient over the covered area by the nozzle flow “\( HTC/A_{nozzle}\) (W/m²-K)/(m²)” against the surface radius which is cooled by the jet water or sprayed water. In the first instance, it is assumed the two types of nozzles (spray and jet) have the same flow (11 GPM) and that they are covering the similar surface areas, so that the ratio of jet nozzles per spray nozzles is 1; it is observed that the spray nozzle (Spray) has a better capacity to remove heat than the jet one (Jet-1-Noz) according to the covered area for the different lengths of the surface radius. As a second case, the flow is reduced by half for the jet nozzle, which means two jet nozzles will be used, and now the area coverage of each jet type nozzle is adjusted in such a way that together the two jet nozzles cover the same area that a single spray nozzle is covering. It is observed how in the “Jet-2-Noz” curve the ratio of “\( HTC/A_{nozzle}\)” improves and the same
surface radius of a spray nozzle still being covered by the jets. As a third case, the initial flow of 11 GPM is now reduced 5 times, so that 5 jet nozzles are placed instead one of spray nozzle "Jet-5-Noz". It is observed that for a smaller surface radius than 0.069 m covered, the “$HTC/A_{nozzle}$” ratio of a jet nozzle is still lower than the spray nozzle’s, but it changes when the distance of the surface radius that is cooled begins to increase. This is because now the spray nozzle must remove heat from a larger area while the jet nozzles are stirring heat for a surface that is 5 times smaller and now the jets have better capacity to remove the heat by the surface area they cover even when they have a lower flow. The last case is now for having 10 jet nozzles per one of spray, they together cover the surface area that a single spray nozzle would do, the flow is also one tenth of the spray for each jet nozzle. It is observed that for all the cooled surface radius, the ratio of “$HTC/A_{nozzle}$” is now greater than the spray nozzle one. This is because each jet nozzle covers a surface area ten times smaller than a single spray nozzle, making that “$HTC/A_{nozzle}$” ratio gets improved.

Figure 4-14. Comparison of jet and spray droplets impingement according to the Heat transfer coefficient per surface area covered.
Previous results show that jet impingement can be as good as the spray droplets impingement for cooling. But one disadvantage is that many jet nozzles are required to cover a similar surface area than a single spray nozzle to have similar heat transfer coefficient ratio per covered area.
Chapter 5. Comparison tubular-panel and spray cooling systems

A fair comparison of the studied cooling systems is performed. The operational conditions of each system are exposed and compared as the inlet pressure, amount of water flow within them, pressure drops, and the power required to overcome them, how the heat removal capacity of each system is affecting the temperatures and energy losses in the walls. A brief discussion about which system operates in safer conditions is presented.

5.1 Pressure, water flow and energy to overcome losses

An approximation for the initial pressure of the piping system is done in the equation (5.1) by an energy balance, to obtain the total water flow of 0.3194 m³/s measured. Assuming that the total flow is discharged to an atmospheric pressure and the main pipe of discharge has a diameter of 0.19726 m, the total head loss of the system ($H_{lp}$) is equal to the piping losses of each panel (35.25 m). For the spray cooling system, the total water flow is the flow obtained for each array nozzle times the 38 arrays to cover the wall; total head loss ($H_{ls}$) is the one of the first nozzles (13.63 m).

$$P_p = \left( \frac{(10.4526 \text{[m/s]})^2}{2 \cdot g} + H_{lp} \right) \cdot \rho_{water} \cdot g$$  

(5.1)

The initial pressure of the piping system obtained with equation (5.1) is 396.958 kPa, 57.6 PSI, approximately 60 PSI.

There is a considerable difference in the pressures used by each system, the table 5-1 shows the pressure that each system operates, the total water flow supplied, and the pressure drop obtained with the equations (5.2) and (5.3).
\[ \Delta P_p = \rho_{water} \times g \times H_{lp} \]  \hspace{1cm} (5.2)

\[ \Delta P_s = \rho_{water} \times g \times H_{ls} \]  \hspace{1cm} (5.3)

Table 5-1. Pressures and total water flow in the cooling systems.

<table>
<thead>
<tr>
<th>Cooling system</th>
<th>Pressure (Pa) / (PSI)</th>
<th>Total water flow (m(^3)/s) / (GPM)</th>
<th>Pressure drop (Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tubular-panel</td>
<td>396958 / 57.6</td>
<td>0.3194 / 5063.85</td>
<td>342797</td>
</tr>
<tr>
<td>Spray nozzles</td>
<td>137895 / 20</td>
<td>0.3125 / 4953.95</td>
<td>132852</td>
</tr>
</tbody>
</table>

Even the tubular-panel has a higher pressure almost 60 PSI, three times bigger than the spray system pressure; the total flow of both systems is very similar, this is because the tubular-panel piping has more losses due to the configuration of panels.

The energy required to overcome the pressure drop/losses in each system is given by the equations (5.4, 5.5) for the tubular-panel and spray systems respectively. In the table 5-2, the power to overcome losses is obtained. Since the tubular-panel system is operating with a higher pressure, therefore a bigger pressure drop, a pump with a higher power is needed.

\[ \dot{W}_p = \dot{V}_o \times \Delta P_p \]  \hspace{1cm} (5.2)

\[ \dot{W}_s = 38 \times \dot{V}_{total} \times \Delta P_s \]  \hspace{1cm} (5.3)

Table 5-2. Energy to overcome losses.

<table>
<thead>
<tr>
<th>Cooling system</th>
<th>Power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tubular-panel</td>
<td>109505</td>
</tr>
<tr>
<td>Spray nozzles</td>
<td>41518</td>
</tr>
</tbody>
</table>

Although the frictional losses or pressure drops were neglected in the piping that connects to each nozzle array (all the nozzle arrays at the same initial pressure), to overcome these losses in practice, around to four or more pumps are required to reduce the frictional losses.
5.2 Heat removal capacity affecting temperatures

In the sections 3.3 and 4.3 the convection heat transfer coefficient that both systems are using to remove the heat was obtained. As the tubular-panel system is operating with a higher pressure and therefore high velocities, the maximum HTC was over the 37 kW/m²-K and the minimum around to 26.5 kW/m²-K for the panels with more losses, as shown in the figure 3-6. The spray cooling system showed lower heat transfer coefficients, which does not mean it is a bad system, the maximum HTC was around to 22 kW/m²-K for the first pair of nozzles in the array, and the minimum around to 16 kW/m²-K as shown in the figure 4-6.

The comparisons of temperatures of both models are for:

- Slag-layer surface temperatures.
- Inner wall surface temperatures (surface behind the slag layer).
- Water temperatures.

**Slag-layer surface temperature.**

The temperature that the slag can reaches, it is very similar for both systems. The figures 3-8 and 4-7 show the temperatures of the slag-layer according to slag thickness and arc coverage when using tubular-panel and spray systems, respectively. There is no distinguishable difference in the temperatures, both systems can get similar slag-layers thickness for the different arc heights, with a maximal thick of 0.045 when the arc is covered.

**Inner wall surface temperature.**

The inner temperature of the cooling systems is shown in the figures 3-9 and 4-8, outer pipe surface temperature for the tubular system and inner plate surface temperature for the spray system, respectively. For the case (A) when the arc has a height of 0.4 m, the maximum temperature of the inner surface was around to 393.2
Chapter 5

K (120 C) for the hottest panels and 412.3 K (139 C) for the hottest zone of the sprayed plate; the difference among the coldest panels is approximately of 5.6 degrees for the tubular system and of 6.6 for the coldest zone of the spray system. For both systems, the temperature tends to be more uniform in panels/zones when the arc height reduces. For the case (C) when the arc is covered, and slag has a thickness of 0.045m, the hottest tubular panel has a temperature around to 326.2 K (53 C) and for the hottest spray zone is around to 329.2 (56 C). As previously seen, the tubular systems have a higher heat removal capacity, therefore the wall panels have lower temperatures than the sprayed plate, almost of 20 degrees of difference when the arc height is 0.4 m. These temperatures do not affect the operation process of the EAF since the steel panels/plates are operating at low temperatures that do not affect their properties.

**Water temperature.**

The final temperatures of the water used in the wall cooling systems are obtained in the figures 3-10 and 4-12, for the tubular and spray systems respectively. The outlet water temperature reaches the 327 K (54 C) for the hottest panels and 318 K (45 C) in the coldest for the case (A) when the arc height is 0.4 m and no slag adheres; 17 and 8 degrees of difference respect to the initial temperature (310 K). For the spray system the final water temperature is obtained from the total heat transferred, since it is a better approximation, the final temperature is around to 326.4 K (53 C), 16.4 degrees of difference respect to initial temperature. For the case (C) when the arc is covered, and the slag is protecting the wall with a thickness of 0.045m, the outlet water temperature is around to 313.4 K (40 C) and 311.8 K (38 C) for hottest and coldest panel respectively; for the final temperature of the sprayed water it is around of 313.3 K (40.15 C). In the case (A) the tubular system has an average temperature of the outlet water around to 323.7 K (50.5 C), 2.7 degrees colder than the one used in the spray system. For the case (C) the average outlet water temperature of panels is around to 312.8 K (39.65 C), just 0.5 degrees colder than the sprayed water. As
the surface temperatures, the water temperature of the spray system tends to get a little warmer because it has a lower convection heat transfer coefficient.

5.3 Energy losses

The heat transfer comparison is done with the heat transfer in the panel 4 respect to the nozzles needed to cover the surface area of the panel. In the figures 3-11 and 4-10 the heat transfer for each panel and sprayed nozzle sections are obtained respectively. The heat removed in the panel 4 when there is no slag and arc height is 0.4 m, it is around to 1.33 MW. For a nozzle section (surface area covered by a nozzle) energy losses are 34795 W, these are almost uniform in all the nozzle sections. To cover the area of the panel 4 with sprayed water, 38.16 nozzles are required, given us a heat transfer of 1.327 MW very similar to the one of panel 4, this is because the cooling system used has a no serious effect on furnace efficiency compared with the arc coverage and slag-thickness conditions that EAF is operating.

The results of figure 4-11 for the total heat transfer when using the spray system, do not consider removing the losses of the slag door, and the total surface covered by the panels; it is assumed that the entire wall is covered by the spray nozzles when using the spray system. In general, very similar results would be obtained for both systems, as shown in the previous case.

From the point of view that the sprayed steel plate has a more regular shape and wedge-shaped elements can be added to retain a higher percentage of slag, known as “C-cup” slag retainers, the slag-layer would tend to be more uniform throughout the wall and with a greater thickness than it would be in the tubular panels, this does that the heat transfer or energy losses of the EAF process decrease significantly and therefore the furnace would be more efficient.
5.4 Safety

For safety operation conditions, the spray systems have an advantage, these operate from the external side of EAF wall, making the water does not pass directly to the molten surface in case of a leak because it is in the form of sprayed droplets and with a very low pressure unlike the tubular systems, which are inside the EAF that in case of a water leak, there is a greater probability that this impacts directly on the molten surface and with very high pressure with the risk of water getting trapped by the molten steel and an steam explosion is generated. Nevertheless, flowmeters can be implemented in the tubular system to detect water leaks and the flow can be controlled by control valves. On the other hand, for spray systems it is more difficult to detect if water is leaking into the furnace where exhaust gas analyzer systems can be used.

5.5 General summary

In the table 5-3, a general summary of the previous comparison of the operating conditions of the EAF, is shown depending on whether the cooling system is the tubular-panel or the spray-nozzles.

<table>
<thead>
<tr>
<th>Cooling System</th>
<th>Tubular-panel</th>
<th>Spray-nozzles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure (PSI)</td>
<td>60</td>
<td>20</td>
</tr>
<tr>
<td>Water flow (GPM)</td>
<td>5063.85</td>
<td>4953.95</td>
</tr>
<tr>
<td>Power to overcome losses (kW)</td>
<td>109.5</td>
<td>41.5</td>
</tr>
<tr>
<td>Heat removal capacity (kW/m²-K)</td>
<td>37-26.5</td>
<td>22-16</td>
</tr>
<tr>
<td>Slag-layer temperature (K)</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Inner wall temperature (K) when</td>
<td>393.2</td>
<td>412.3</td>
</tr>
<tr>
<td>harc=0.4 m – 0 m</td>
<td>326.2</td>
<td>329.2</td>
</tr>
<tr>
<td>Water temperature (K) when</td>
<td>323.7</td>
<td>326.4</td>
</tr>
<tr>
<td>harc=0.4 m – 0 m</td>
<td>312.8</td>
<td>313.3</td>
</tr>
<tr>
<td>Energy losses</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Safety</td>
<td>x</td>
<td>✓</td>
</tr>
</tbody>
</table>
Summarizing results of table 5-3, the tubular-panel system needs an inlet pressure about 3 times higher than the one used for the spray-nozzles system. Both systems operate with similar total water flows. High pressures represent more required power to overcome the losses in cooling systems, therefore a greater power is needed in the tubular-panel. The tubular-panel system has a greater capacity to remove heat, therefore the wall and fluid temperatures are lower than the spray system. The energy losses are not affected by the cooling systems, both present similar values. Spray systems have the advantage of operating from the outside of EAF, making them more reliable for a safer EAF operation.
Chapter 6. Conclusions

A heat transfer study was carried out on the walls of an electric arc furnace using two types of cooling systems, tubular-panel and spray-nozzles. Assuming only the radiation of the electric arc and the molten surface as the source of heat energy that heats the walls of the furnace. Mathematical models were developed to find the water flow distribution in the cooling systems and their capacities to remove the heat, obtaining the heat transfer and temperatures in the walls.

The temperatures and energy losses in the EAF wall by the cooling systems were verified to be lower when the arc is covered and when a large amount of slag adheres to them. Comparing the cases when the arc has a height of 0.4 m without slag adhered on the walls against the arc completely covered with a maximal slag-layer of 4.5 cm on the wall, the energy losses were reduced up to five times and the inner temperatures of the wall were reduced more than 70 degrees.

Extended surfaces or slag retainers that are added to the steel plate or panels inside the furnace, are critical for a thick layer of slag to form. It was found that the slag-layer on the walls plays a very important role in the steel-making process, makes the furnace operate in a more efficient manner and maintains the surface and water temperatures in the panels at lower levels, thanks to its low thermal conductivity. Slag works as an insulation and heat inside the furnace is better used to produce steel.

The heat-removal capacity and temperatures of walls depend highly of the water flow, which was found to differ for each panel and sprayed zone of the EAF wall. In the tubular-panel system, there exist a difference of water flow around 3%, respect to the total flow, for the panels with the lower flow against those with higher flow. In the spray system, the difference of water flow is around 2.8%, respect to the total
flow in the array of nozzles, for the nozzles with the lower flow against the first nozzles of the array.

For the best EAF-operation scenario, where the maximal slag thickness adheres to the panels and the arc is completely covered, the losses are around to 2.4%-2.9% for an average arc power of 140 MW and the outlet water temperatures differ by 2 or 3 degrees from the inlet water temperature for the both systems.

With the tubular-panels models, the slag-layer thickness in each panel can be estimated by knowing the outlet water temperature, as well as the arc coverage and the amount of energy in MW that is lost by the cooling system, which can be used or saved in the process. This study identifies the panels with the highest temperatures and heat rates, meaning it can be used to perform predictive maintenance to avoid steam explosions. It is also observed that some panels have greater flow rates than others; this flow can be redirected to the panels with lower capacities to remove heat by a flow control and thus maintain a uniform temperature throughout the wall.

For the spray cooling system, were found the hottest sections of the wall to those where nozzles are near to the lower part of the wall, since these nozzles are the ones with lower percentage of flow. The upper part of the wall is the one that operates with lower temperatures, since the first nozzles of the array are the ones that take the highest percentage of flow.

It was verified spray droplets impingement is a better choice for cooling than the jet impingement, since to perform similar results of the spray system, a larger number of jet nozzles are needed to cover the same surface area, therefore a higher pressure will be needed to overcome losses in pipes.

Both systems operate with acceptable operating temperatures on their walls, which would not affect the material properties or the steel-making process.
A fair comparison was done for both systems. The tubular-panel system has a higher heat removal capacity than the spray system and therefore lower inner surface and final water temperatures. Nevertheless, the tubular-panel system operates with a higher pressure almost three times bigger than the spray system therefore, more energy is needed to overcome the friction losses. It was proved that the energy losses of the process do not depend on the cooling system implemented, it does on the conditions of arc coverage and thickness of the slag-layer adhered on the wall. However, the spray system has a greater uniformity to remove the heat of walls. Spray systems are more reliable for safety operating conditions since the cooling is done from the outside with a much lower probability of a steam water explosion occurring in the event of a leak.

This study presented practical methods for estimating the temperatures and heat transfer inside the EAFs according to the water-flow distribution in the panel piping of the tubular system and to the mean volumetric flux on water sprayed surfaces.

### 6.1 Recommendations for future works

In order to complement the research carried out, the following is recommended:

- The pressure drops in the main piping which injects the flow to the panels and to the nozzle arrays should be considered.
- Obtain experimentally and by simulation the losses coefficient of nozzles and mean diameter of droplets.
- It is necessary to know how the arc power influences its temperature and its geometry.
- View factors should be obtained from the current form of the EAF and electric arc.
• Consider the variation of the slag-layer thickness along each panel, since in this study it was considered to be uniform and with an average thickness along the entire panel.

• Investigate how heat transfer is affected by the effect of multiple nozzles.

• It is recommended to include in this work, the heat emitted by the gases inside the furnace and the heat removed by the surroundings and other elements that were neglected in this study.

• With the models proposed in this thesis, a simple engineering-control system can be implanted to maintain a uniform surface temperatures and outlet water temperatures in the panels, by regulating the water flow in the panels by means of control valves and sensors.

• A computational fluid dynamics simulation of heat transfer is recommended for the two EAF cooling systems studied, to verify the results shown in this study.
Nomenclature

\( A_{ai/bi} \) Surface area covered by each spray nozzle, nozzle section
\( A_{\text{Intpipe}} \) Internal cross-sectional area of pipes
\( A_{\text{nozzle}} \) Surface area covered by a nozzle
\( A_pn \) Surface area of each panel
\( A_{\text{roof}} \) Surface area of roof
\( A_{s1} \) Molten surface area
\( A_{\text{sarc}} \) Surface area of the arc
\( A_{\text{surr}} \) Surface area of surroundings
\( A_w \) Total wall-surface area
\( Cp \) Specific heat of liquid water
\( d_{arc} \) Arc diameter
\( d_{32_{ai/bi}} (d_{32_{ai}}, d_{32_{bi}}) \) Sauter mean diameter of the droplets impacting each nozzle section
\( d_i \) Secondary pipes diameter
\( d_o \) Diameter of the main pipe
\( d_{or} \) Diameter of the nozzle orifice
\( d_{\text{pipe}} \) Internal pipe diameter
\( e \) Pipe roughness
\( F_{1-\text{roof}} \) View factor from molten surface to roof
\( F_{12} \) View factor from molten surface to walls
\( F_{12_{ai/bi}} (F_{12_{ai}}, F_{12_{bi}}) \) View factor from molten surface to each nozzle section
\( F_{12pn} \) View factor from molten surface to each panel
\( f_{ai/bi} (f_{ai}, f_{bi}) \) Friction factor of each section that connects to a nozzle
\( F_{\text{arc}-1} \) View factor from arc to molten surface
\( F_{\text{arc}_{ai/bi}} (F_{\text{arc}_{ai}}, F_{\text{arc}_{bi}}) \) View factor from arc surface to each nozzle section
\( F_{\text{arc}_{pn}} \) View factor from the arc to each panel
\( F_{\text{arc}_{surr}} \) View factor from arc to surroundings
\( F_{\text{arc}_{wall}} \) View factor from arc to walls.
\( f_{oi} \) Friction factor of each section of the main pipe
\( f_{pn} \) Friction factor of each panel
\( g \) Acceleration due to gravity
\( H \) Separation of nozzle to surface impacted.
\( h_{ai/bi} (h_{ai}, h_{bi}) \) External heat transfer coefficient of each nozzle section
\( h_{arc} \) Arc height
\( h_{jet} \) Heat transfer coefficient of water jet.
\( H_{Lai/Lbi} (H_{Lai}, H_{Lbi}) \) Head losses of the pipe section that connects to each nozzle
\( H_{LP} \) Total head losses of the tubular-panel system
\( h_{Lpn} \) Head losses of each panel
\( H_{LS} \) Total head losses of the spray system
\( h_{pn} \) Internal-heat-transfer coefficient of each panel
\( h_{wall} \) Walls height
\( K_{Lcpn} \) Loss coefficient for bends and curvatures in each panel
\( K_{Lpn} \) Minor loss coefficient of each panel
\( K_{LVpn} \) Loss coefficient of valves in each panel
\( K_{nozzle} \) Loss coefficient of the nozzles
\( K_{slag} \) Thermal conductivity of the slag
\( K_{steel} \) Thermal conductivity of the steel
\( K_{TL} \) Loss coefficient for line flow
\( K_{water} \) Thermal conductivity of the water
\( L_i \) Secondary pipes length
\( L_{oi} \) Separation distance between each nozzle
\( L_{plate} \) Thickness of the steel plate
\( L_{pn} \) Wet length of each panel
\( L_{slagai/bi} (L_{slagai}, L_{slagbi}) \) Slag-layer thickness in each nozzle section
\( L_{slagpn} \) Slag-layer thickness in each panel
\( L_{steel} \) Thickness of the Steel pipe
\( \dot{m}_{ai/bi} (\dot{m}_{ai}, \dot{m}_{bi}) \) Mass flow crossing through in each nozzle section
\[ \dot{m}_{pn} \] Mass flow in each Panel

\[ \dot{m}_{total} \] Total mass flow of the spray system

\[ Nusselt_{al/bi} (Nusselt_{al}, Nusselt_{bi}) \] Estes and Mudawar Nusselt number in the surface of each nozzle section

\[ Nusselt_{jet} \] Nusselt number of jet inpigment

\[ NusseltR_{al/bi} (NusseltR_{al}, NusseltR_{bi}) \] Rybicki and Mudawar Nusselt number in the surface of each nozzle section

\[ P_0 \] Initial Pressure of nozzles array

\[ P_p \] Initial Pressure of the tubular-panel system

\[ Pr_{water} \] Water Prandtl number

\[ \dot{Q}_{al/bi} (\dot{Q}_{ai}, \dot{Q}_{bi}) \] Heat transfer in each nozzle section

\[ \dot{Q}_{pn} \] Heat transfer in each panel

\[ \dot{Q}_{Total} \] Total heat transfer in the walls for tubular-panel system

\[ \dot{Q}_{Totals} \] Total heat transfer in the walls for spray cooling system

\[ R_{cond_{slagpn}} \] Thermal resistance by slag conduction of each panel

\[ R_{cond_{steelpn}} \] Thermal resistance by steel conduction of each panel

\[ R_{conv_{fluidpn}} \] Thermal resistance by water convection of each panel

\[ Re_{al/bi} (Re_{al}, Re_{bi}) \] Reynolds number in the pipe section that connects to each nozzle

\[ Re_{jet} \] Reynolds number of water jet

\[ Re_{oai/obi} (Re_{oai}, Re_{obi}) \] Reynolds number based on nozzle orifice diameter

\[ Re_{oi} \] Reynolds number in each section of the main pipe

\[ Re_{pn} \] Reynolds number in each panel

\[ Re_{sai/sbi} (Re_{sai}, Re_{sbi}) \] Reynolds number in the surface of each nozzle section

\[ r_{int} \] Internal radius of EAF

\[ R_{slag_{al/bi}} (R_{slag_{al}}, R_{slag_{bi}}) \] Thermal resistance by slag conduction of each nozzle section

\[ R_{steel_{al/bi}} (R_{steel_{al}}, R_{steel_{bi}}) \] Thermal resistance by steel conduction of each nozzle section
\( R_{total_{ai/bi}} (R_{total_{al}}, R_{total_{bi}}) \) Total thermal resistance of each nozzle section

\( R_{totalpn} \) Total thermal resistance of each panel

\( r_{sj} \) Radius of the impacted area by the water jet.

\( T_{arc} \) Surface temperature of the arc

\( T_{fin} \) Inlet water temperature

\( T_{fout_{sw}} \) Final water temperature of the sprayed water

\( T_{foutpn} \) Outlet water temperature in each panel

\( T_{s1} \) Molten surface temperature

\( T_{s2_{ai/bi}} (T_{s2_{al}}, T_{s2_{bi}}) \) Slag-layer surface temperature of each nozzle section

\( T_{s2pn} \) Slag-layer surface temperature of each panel

\( T_{s3_{ai/bi}} (T_{s3_{al}}, T_{s3_{bi}}) \) Inner surface temperature of the plate of each nozzle section

\( T_{s3pn} \) Outer pipe surface temperature of each panel

\( T_{s4_{ai/bi}} (T_{s4_{al}}, T_{s4_{bi}}) \) Outer plate surface temperature of each nozzle section

\( U_{ai/bi} (U_{ai}, U_{bi}) \) Global heat-transfer coefficient of each nozzle section

\( U_{pn} \) Global heat-transfer coefficient of each panel

\( V_{ai/bi} (V_{ai}, V_{bi}) \) Velocity of water in the pipe section that connects to each nozzle

\( \dot{V}_{ai/bi} (\dot{V}_{ai}, \dot{V}_{bi}) \) Volumetric flow in the pipe section that connects to each nozzle

\( \bar{V}_{ai/bi} (\bar{V}_{ai}, \bar{V}_{bi}) \) Mean volumetric flux in each nozzle section

\( V_{jet} \) Velocity of the water jet

\( \dot{V}_{o} \) Initial volumetric flow of the tubular-panel system

\( V_{oi} \) Velocity of water in each section of the main pipe

\( \dot{V}_{oi} \) Volumetric flow in each section of the main pipe

\( V_{pn} \) Fluid velocity in each panel

\( \dot{V}_{pn} \) Volumetric flow of each panel

\( \dot{V}_{total} \) Total volumetric flow in a single array of nozzles

\( W_{e_{obi}} \) Weber number based on nozzle orifice diameter

\( W_{p} \) Energy to overcome losses in the tubular-panel system
\[ \dot{W}_s \quad \text{Energy to overcome losses in the spray system} \]
\[ z_i \quad \text{Height of each nozzle} \]
\[ z_o \quad \text{Height of the water supply} \]
\[ \alpha \quad \text{Fully developed turbulent flow correction factor} \]
\[ \beta \quad \text{Ratio between nozzle orifice diameter and inlet diameter} \]
\[ \Delta P_{ai/\text{bi}} (\Delta P_{ai}, \Delta P_{bi}) \quad \text{Pressure drop of each nozzle} \]
\[ \Delta P_p \quad \text{Pressure drop in the tubular-panel system} \]
\[ \Delta P_s \quad \text{Pressure drop in the spray system} \]
\[ \varepsilon_1 \quad \text{Emissivity of the slag} \]
\[ \varepsilon_{\text{arc}} \quad \text{Emissivity of the arc} \]
\[ \mu_{\text{water}} \quad \text{Water dynamic viscosity} \]
\[ \rho_{\text{gas}} \quad \text{Density of the steam water} \]
\[ \rho_{\text{water}} \quad \text{Water density} \]
\[ \sigma \quad \text{Stefan-Boltzmann constant} \]
\[ \sigma_{\text{water}} \quad \text{Water surface tension} \]
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Curriculum Vitae

Josué Contreras Serna was born in Monterrey, México, on November 16, 1993. He earned the Mechatronics Engineering degree from the Universidad Autónoma de Nuevo León (UANL), in June 2015.

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