DESIGN OF AN AIR-MIST ATOMIZER FOR SURFACE COOLING PURPOSE

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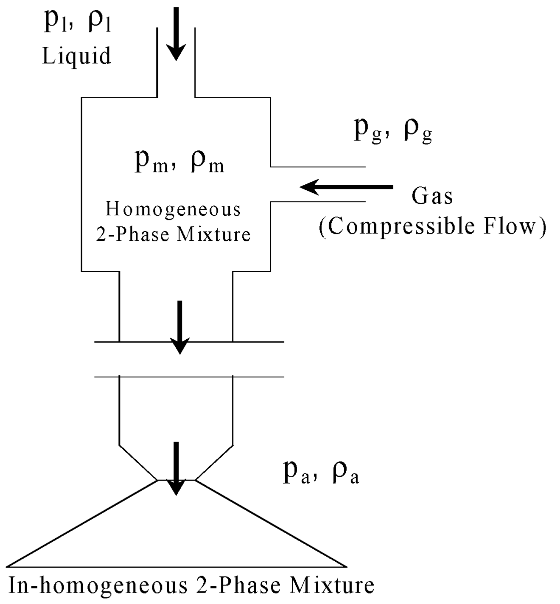
Surface cooling in the steel industry requires a wide operation range of the nozzle flowrate. High water control ratios can be achieved with air-mist nozzles of the internal mixing types. An attempt is presented to simulate the control behavior of the nozzle with a physical model using Bernoulli's law for incompressible (water), compressible (gas) and two-phase fluids (air-water mixture). Comparison of computation with measured values shows, that the air and water flow into the mixing chamber is not only influenced by the pressure drop across the inlet bores, but also by the volume fraction of the liquid phase in the two-phase mixture. A correction for the flow resistance of the air channel was necessary in the computer model in order to fit the measured values. The energy consumption of the nozzle can be calculated as a result of air and water flow and allows estimating the operation costs. Good mixing and pre-atomization of air and water is essential for the required water distribution quality, including the overlapping areas of adjacent nozzles. Air-mist nozzles of the internal mixing type show large variations of the droplet size. The efficiency of spray cooling can be described by the heat transfer coefficient which can be measured under laboratory conditions. Cooling under continuous casting conditions, i.e. in the range above the Leidenfrost point, depends on the energy of the spray impinging on the hot surface.

1. Introduction

Surface cooling, for example in continuous casting applications in the steel industry, requires a wide operation range of the nozzle flowrate in order to cover the need for all production tasks i.e.: casting of different slab sizes and materials at different casting speeds. High water control ratios at small pressure variations can be achieved only with spillback nozzles or with air-mist nozzles of the internal mixing type. The latter are commonly used in steel production, and atomizing water with compressed air allows a wider band of heat transfer coefficients to be covered than with hydraulic nozzles. Air-mist flat jet nozzles have other advantages over flat jet hydraulic nozzles: they cover a wider spray thickness, thus applying the water to a larger surface, and they have small droplets at low liquid output condition and large droplets a high flowrates. This allows an even distribution of the water, even at low impingement density. Cooling with air-mist nozzles results generally in higher heat transfer coefficients in the range above the Leidenfrost point due to better penetration of the liquid film by the air-water two-phase mixture. But operation costs of these air-mist nozzles are higher than that of hydraulic nozzles.



P_l = liquid pressure
P_g = gas pressure
ρ_l = liquid density
ρ_g = gas density
P_m= pressure of mixture
ρ_m = density of mixture
P_a = ambient pressure
ρ_a = ambient density

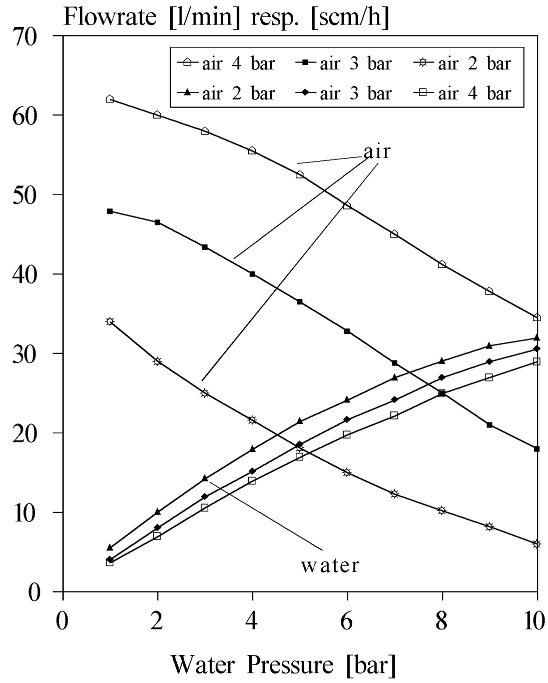


Figure 1 Flow parameters of two-phase mixture and flowrate diagram of the air-mist nozzle

2. Model for the Fluid Flow in an Air-Mist Nozzle

In air-mist atomizers with internal mixing the pressure in the mixing chamber can be controlled by increase or decrease of the gas feed pressure. This allows controlling the water flow in a wide range by variation of the pressure drop in the water inlet channel. In this paper an attempt is presented to simulate the control behavior of this type of nozzle with a simple physical model (Figure 1), using Bernoulli's law for incompressible (water), compressible (gas) and two-phase fluids (air-water mixture). All calculations and measurements in this investigation have been done on the base of air and water.

The flow of the liquid into the mixing chamber can be described by Bernoulli's law (1) for incompressible fluids, where c_l is the contraction coefficient of the liquid inlet bore, A_l its cross section area and V_l the liquid volume:

$$p_l - p_m = \frac{\rho_l}{2} \cdot \frac{\dot{V}_l^2}{c_l^2 A_l^2} \qquad (1)$$

The gas flow into the mixing chamber is compressible and, thus, obeys the law (2) with the gas velocity v_g and the adiabatic exponent κ.

$$v_g = \sqrt{2 \cdot \frac{\kappa}{\kappa - 1} \cdot \frac{p_g}{\rho_{g,1}} \left[1 - \left(\frac{p_m}{p_g} \right)^{\frac{\kappa - 1}{\kappa}} \right]} \qquad (2)$$

The flow of the two-phase mixture out of the mixing chamber is treated as homogenous flow and can be described by the generalized law of Bernoulli (3) for two-phase mixtures with v_m as mean velocity of the two phases and α the volume fraction of the liquid phase [1]:

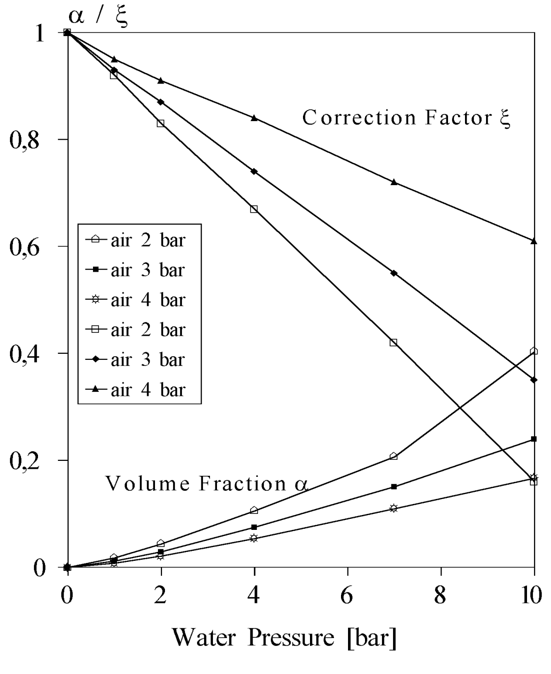
$$p_a + \frac{r_l}{2} \cdot v_m^2 = p_m \left(1 + \frac{1-a}{a} \cdot \ln \frac{p_m}{p_a} \right) \qquad (3)$$

The contraction coefficients had been determined by flow measurements on single nozzle components, i.e. flow of water or air through the water or air inlet orifices respectively the nozzle outlet opening. The parameters shown in Table 1 have been used.

Table 1 Parameters of nozzle inlet and outlet openings

	Orifice diameter [mm]	Contraction coefficient
Liquid c_l	4,1	0,9865
Gas c_g	5,2	0.77
Mixture c_m	9,54	0,81

Comparison of computation with measured values, as shown in Figure 4 and 5, confirm, that the air and water flow into the mixing chamber is not only influenced by the pressure drop across the inlet bores, but also by the volume fraction α of the liquid phase (water) in the two-phase mixture. A greater percentage of water in the mixture causes an occultation effect for the airflow and increase the flow resistance of the air inlet channel, thus simulating a decrease of the inlet diameter with increasing percentage α of the water volume in the mixing chamber. This allows further increase of the control range of the nozzle.



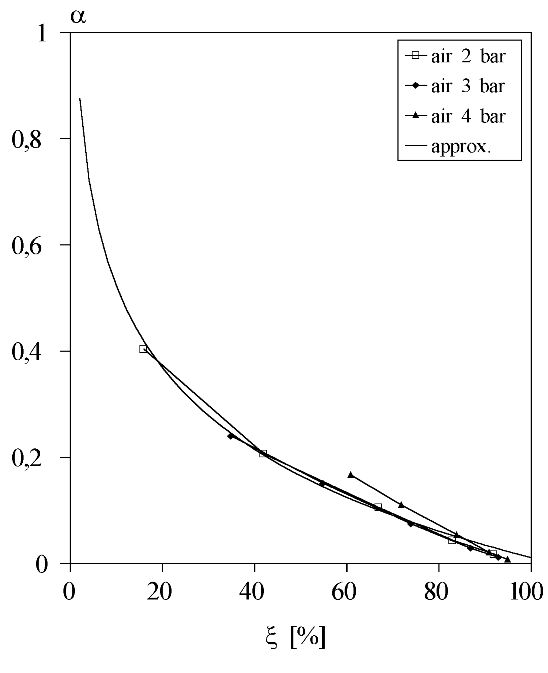
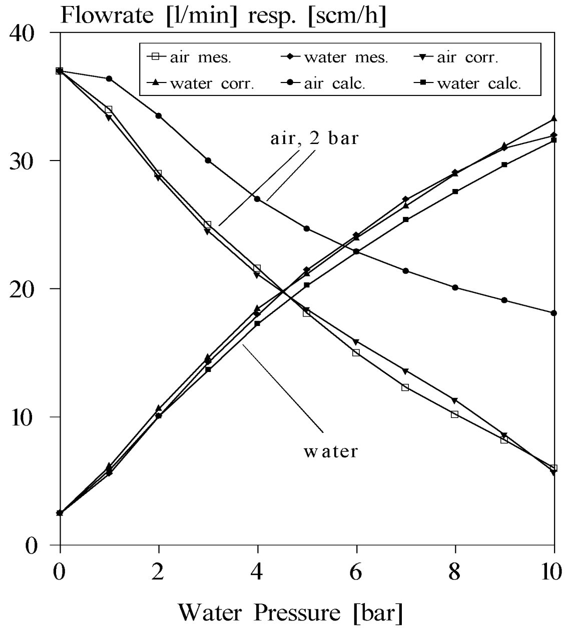
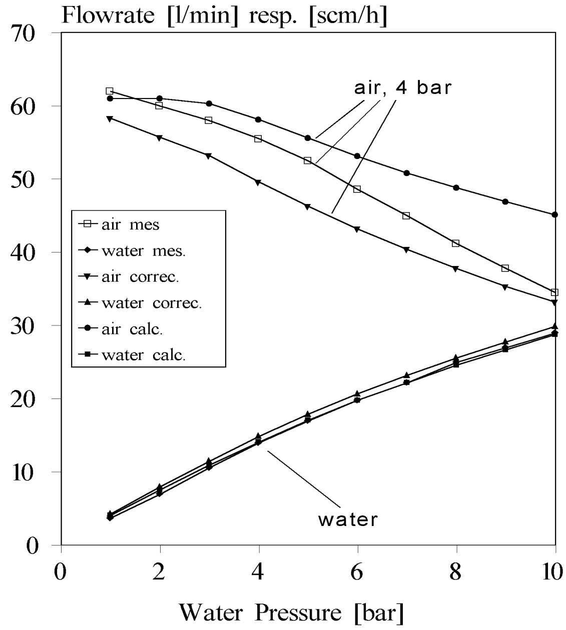


Figure 2 Volume fraction α and correction factor ξ **Figure 3** Volume fraction α vs. correction factor ξ

A correction factor ξ for the flow resistance of the air channel as a function of α was necessary in the computer model in order to fit the measured values. With this factor the effective area of the gas inlet is $A_g' = \xi \cdot c_g \cdot A_g$. Figure 2 shows α and the correction of the gas inlet orifice as a function of the water and the air pressure, and Figure 3 shows the correction factor ξ versus α.

Theoretical calculations and measurements show that a high control rate of air-mist nozzles with internal mixing is achieved with a small outlet orifice of the nozzle and small dimension of the mixing chamber on the one hand, and large inlet bores for both, air and liquid, on the other hand. But these nozzles react very sensitive to fluctuations of the feed pressure and need precise pressure control instrumentation.

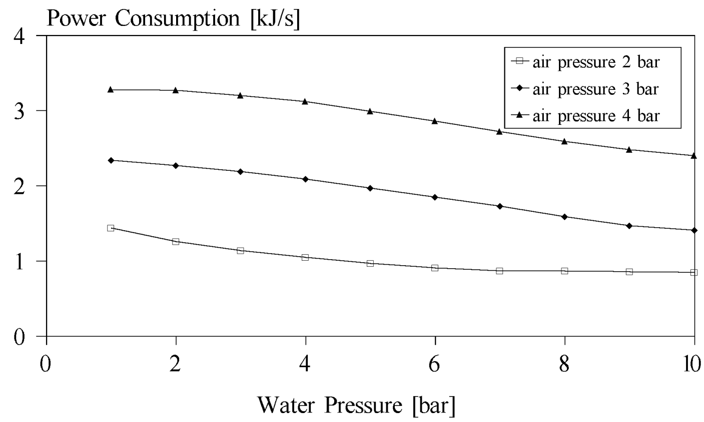




air mes. = measured air flowrate; **water mes.** = measured water flowrate;
air calc. = calculated air flowrate; **water calc.** = calculated water flowrate;
air corr. = calculated and corrected air flowrate; **water corr.** = calculated water flowrate with corrected air flowrate

Figure 4 Measured and computed flowrates, 2 bar **Figure 5** Measured and computed flowrates, 4 bar

The flowrates at given feed pressures allow to compute the energy consumption of the nozzle with equation (4). Figure 6 shows the data for the nozzle, studied in this investigation. The total energy consumption of the nozzle is a result of air and water flow [2] and allows estimating the operation costs of the nozzle. It can be shown that low water flowrates result in higher energy values due to the increase of airflow, i.e. the wide control range of air-mist nozzles with internal mixing is achieved by the energy of the compressed air. This can be compensated to a certain extent by using low air pressures at low water flow rates.



Specific power consumption of the nozzle:

$$\frac{P_{tot}}{\dot{m}_l} = \frac{p_l}{r_l} + \frac{\dot{V}_g}{\dot{m}_l} \cdot \frac{r_g \cdot R \cdot T}{M_g} \cdot \ln\frac{p_g}{p_m} \quad \textbf{(4)}$$

P_{tot} = total power consumption
m_l = mass of the liquid
M_g = molar mass of the gas
R = universal gas constant

Figure 6 Power consumption of air-mist nozzle with internal mixing

3. Water Distribution Characteristic

Though droplet size does not have a significant influence of the cooling efficiency, a good mixing and pre-atomization of air and water is essential for the required water distribution quality. The steel to be cooled must be covered evenly with water under all conditions, and this applies also to the overlapping areas of adjacent nozzles. Different designs for the mixing of water and air in the mixing chamber have been developed in order to get a homogenous two-phase mixture in the nozzle chamber, which is an assumption for creating a well-defined spray pattern with the nozzle tip. Though variations of the heat transfer are not proportional to the impinging liquid flow, the water distribution is regarded as a critical parameter of air-mist nozzles for surface cooling purposes, and it is a subject of special quality control procedures.

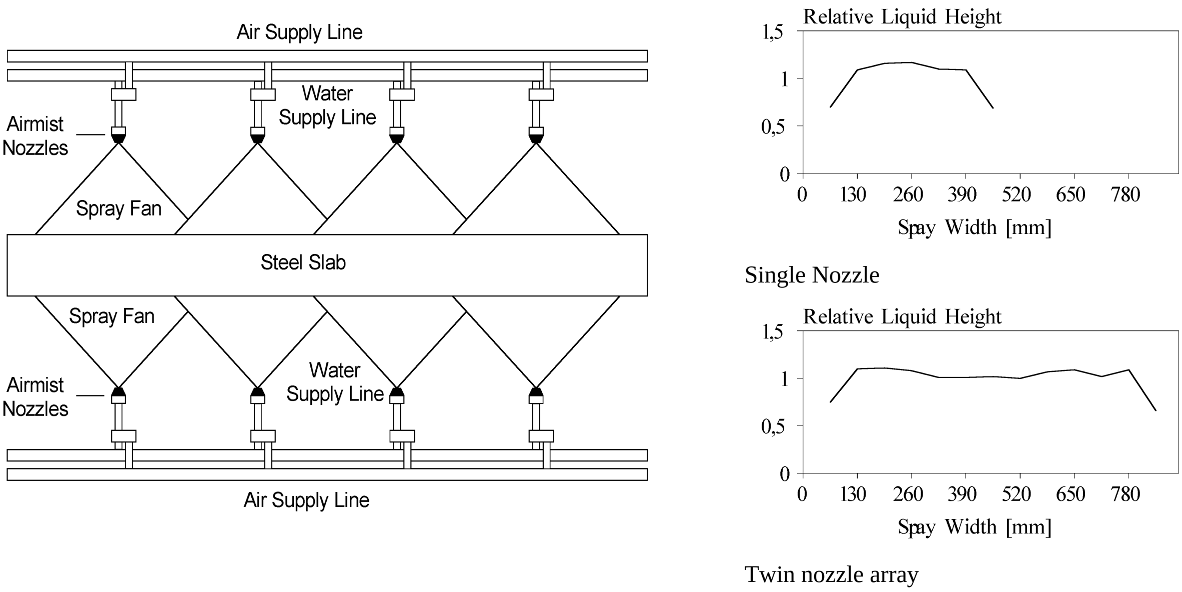


Figure 7 Arrangement and liquid distribution of Mastercooler nozzles for continuous casting

4. Droplet Size and Velocity

Droplet size analysis has been done with a laser-phase-doppler-analyser, measuring simultaneously droplet size and velocity in a distance of 300 mm from the nozzle outlet . Figure 8 shows the Sauter mean diameter for two different air pressures versus the water pressure. The droplet diameter D_{32} covers the range from 150 to 680 µm, and this large variation results in a significant change of the number of droplets. With this number being proportional to ~ $1/D^3$ (where D represents any characteristic diameter), the density of water particles, falling on the hot surface, increases with decreasing water pressure, despite the fact that the water flowrate covers a ratio of about 1:10 over the full pressure range. Though the heat transfer of the nozzle is not a direct function of the droplet size, the advantage of an air-mist nozzle over an hydraulic version can be seen clearly: With decreasing water flowrate, i.e. decreasing water impingement density, the even coverage of the surface to be cooled is maintained by the increasing number of droplets and their decreasing diameter.

The mean droplet velocity is shown in figure 9. This parameter depends to a great extent on the velocity of the atomizing air. Especially at 2 bar air pressure, the droplet velocity is more or less constant, while at 4 bar air pressure there is an increase by a factor 2,2. Because the heat transfer coefficient is a function of the kinetic energy of the water, impinging on the surface [3], this increase in droplet velocity explains an increase in the heat transfer with increasing air pressure.

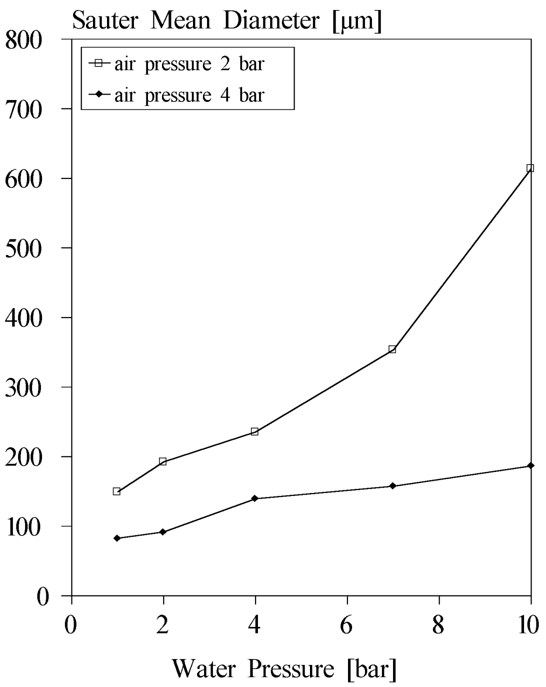


Figure 8 Droplet size of the air-mist nozzle

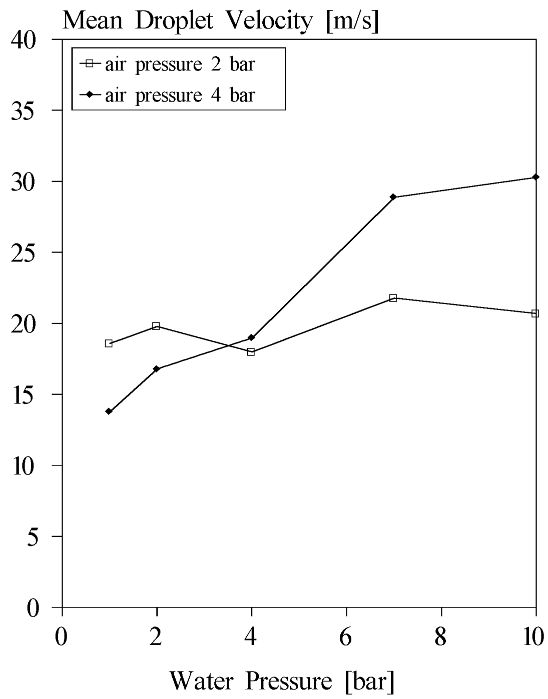


Figure 9 Mean velocity of droplets

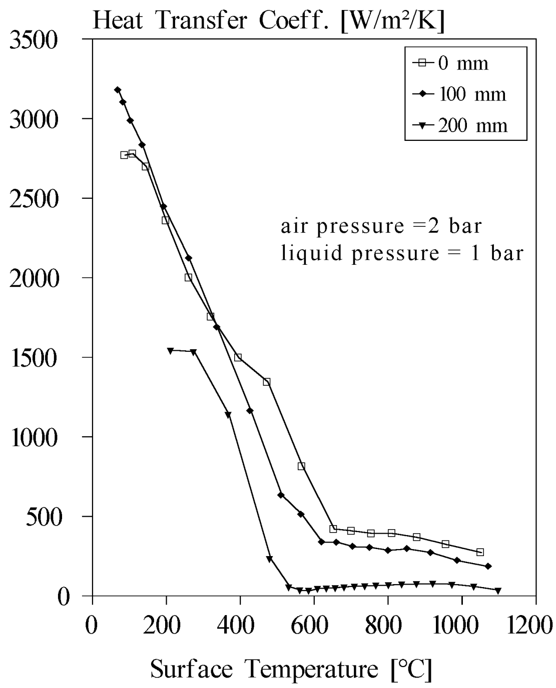
5. Heat Transfer Coefficient of the Air-Mist Nozzle

Heat transfer of spray nozzles today can be measured with laboratory test benches, which simulate the conditions in the continuous casting plant (moving nozzle test, [4], [5]). Figures 10 – 13 show results of such tests, made with a spray distance of 300 mm to the surface to be cooled. The surface, a hot steel plate, was moved forward and back with a speed of 1 m/min. Cooling experiments were done for initial temperatures of 1250°C. The internal temperatures are measured and surface temperature and heat transfer coefficient distribution is computed by the inverse task.

Figure 10 shows the heat transfer coefficient versus the surface temperature for 0, 100 and 200 mm distance from the nozzle axis, averaged over the direction of motion in a range of –200 to +200 mm. It can be seen clearly, that at low liquid and air pressures there is a well-defined Leidenfrost effect. This results in a – more of less – constant heat transfer coefficient at high surface temperatures. Figure 11, however, does not show such an effect. Due to high kinetic energy of the spray, the boiling liquid film on the hot surface is penetrated and there is no Leidenfrost phenomenon in the temperature range up to 1100°C. The Leidenfrost temperature will occur at higher temperatures.

Figure 12 and 13 show values for the heat transfer coefficient, averaged over the total spray area, for 700°C and 1000°C versus the water pressure for two different air pressures. The figures show the control range of the heat transfer coefficient of the nozzle when operated in a water and airflow range as shown in figure 1. It can be seen, that not only higher water pressure but also higher air pressure result in an increase of the heat transfer. But this is also due to the fact, that the Leidenfrost point is shifted up to higher temperatures with increasing water and air pressures and flowrates, i.e. increasing kinetic energy hitting the surface.

Figure 14 shows the influence of the kinetic energy of the water, impinging on the surface to be cooled, on the mean heat transfer coefficient. As has been demonstrated in [3], the kinetic energy of the spray seems to be to most dominant factor for the efficiency of spray cooling. A



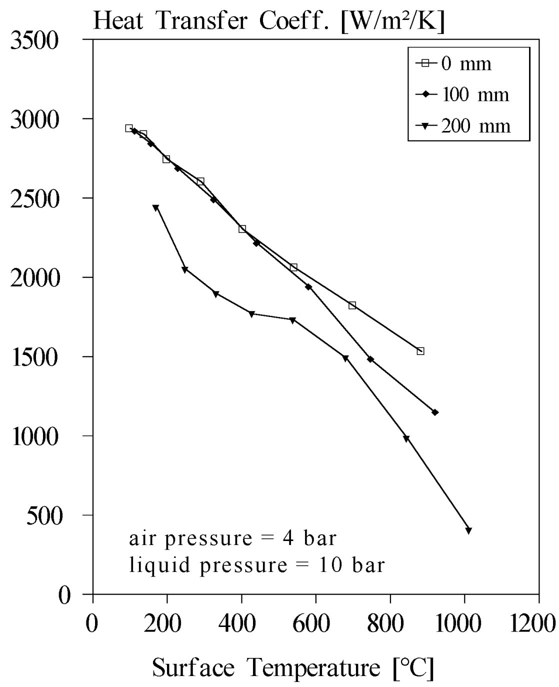
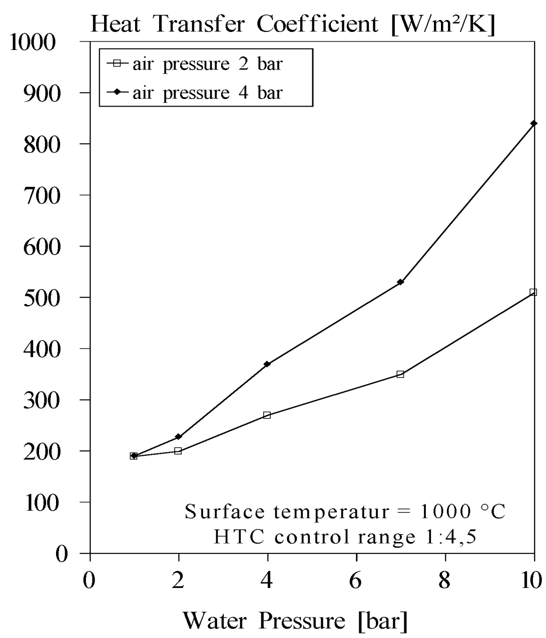


Figure 10 Heat transfer coefficient for 2/1 bar

Figure 11 Heat transfer coefficient for 4/10 bar

slight difference between the lines for air pressures of 2 bars and 4 bars can be seen in Figure 14. Particularly at higher air pressures, higher heat transfer coefficients can be achieved, because the 2-phase mixture of water and air results in higher values for E_{kin} than at lower air pressures. In other words, the additional energy of the compressed air, which is used for atomization, increases the kinetic energy of the impinging water and, thus, the heat transfer.



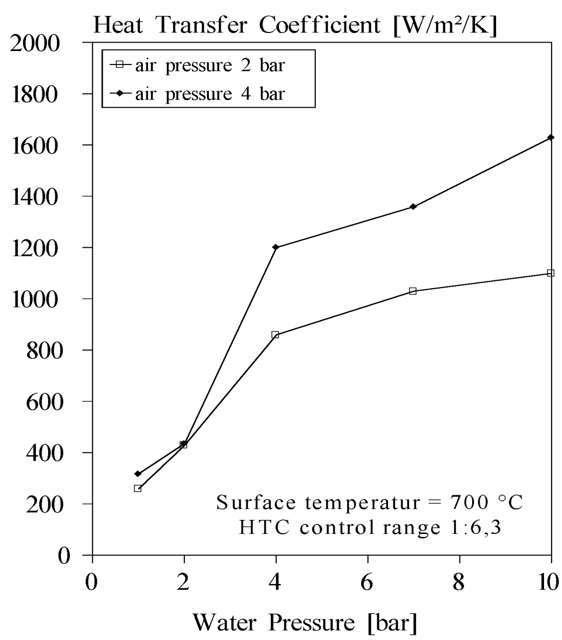
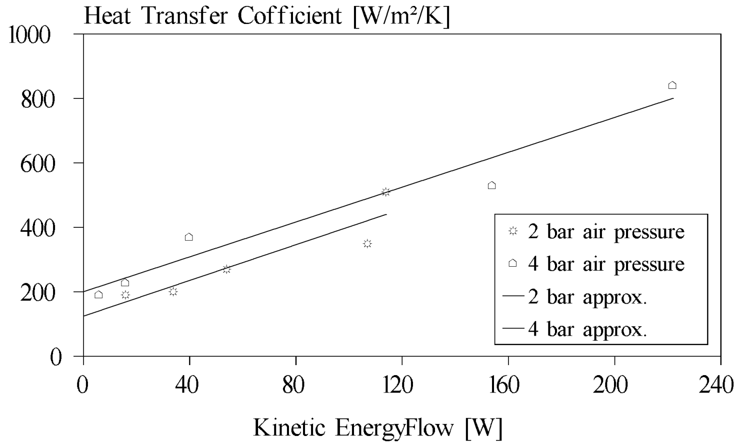


Figure 12 Mean heat transfer coefficient at 1000°C

Figure 13 Mean heat transfer coefficient at 700°C



The total water mass flow of the nozzle and the mean velocity of the water drops define the kinetic energy flow per time unit here.

$$\dot{E}_{kin} = \frac{r \cdot \dot{V}}{2} \cdot v_{mean}^2$$

The dimension is J / s.

Figure 14 Mean heat transfer vs. kinetic energy of the water impinging on the surface

6. Conclusions

Air-mist nozzles of the internal mixing type have been proved at the best solution for spray cooling in continuous casting applications. The control behavior can be estimated with a simple physical model which allows calculating the air and water flowrates at different pressures and given geometrical properties of the inlet and outlet bores (diameter, shape). A large water control range is achieved with large inlet and small outlet bores. The droplet size spectrum varies over a large range, due the wide water control range, which results in a large variation of the air to liquid ratio. This allows an even coverage of the surface to be cooled, because the number of drops increases with decreasing water flowrate. The more critical part of the nozzle design is the requirement to guarantee an even water distribution under all operation conditions, with single as well as with overlapping nozzles. Heat transfer measurements show, that the heat transfer coefficient at high temperature is influenced by the Leidenfrost phenomenon. The Leidenfrost temperature increases with increasing water and air pressure in the air-mist nozzle. This is the reason for the large range of heat transfer coefficient, which can be covered by this type of nozzle.

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