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# Research on Cooling Efficiencies of Water, Emulsions and Oil

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Pure water is considered as a coolant in many technologies connected with steel processing. This presumption is not quite true because plant water contains various pollutants. Some technologies use additives for lubrication and anti-corrosion reasons. A typical example is in the cold rolling of steel, where water based emulsions are used for the cooling of rolls and for lubrication. In hot rolling technology, oil is now frequently used for the lubrication of rolling gaps. Part of this oil sticks to the roll surface and mixes with cooling water. Pure oil is used as a coolant in rolling of some non-ferrous metals, while other metallurgical processes use water, but water becomes polluted by oil and contains a higher mineral salt content. The spray cooling efficiency of these coolants is different from the cooling efficiency of pure water. This study showed that even a small percentage of an additive can significantly reduce the cooling ability of water. Cooling intensity achieved in plant conditions can be less, especially with an increase of pollutant concentration in the cooling circle. The study presented here compares spray cooling with pure water to the cooling achieved using water-based oil emulsions of various concentrations and cooling using pure oil.

Keywords: SPRAY COOLING, WATER COOLING, EMULSION COOLING, OIL COOLING, EXPERIMENTAL

## Introduction

Pure water is a coolant very rarely used in the metallurgical industry. Plant cooling water contains some undesirable pollutants and some desirable additives. These additives are amongst other reasons; frequently for lubrication, sometimes for prevention of corrosion and for the suppression of biological activity of coolants. In some technologies, such as aluminium rolling, emulsions with a high percentage of oil or pure oil are used. In most of the experimental studies of cooling, pure water is used. This study compares the differences in cooling ability of real coolant versus pure water. The cooling effectiveness of oil and water emulsion is compared to water cooling effectiveness. Results were obtained on experiments using spray cooling. The range of temperatures used was intended to cover the cooling of rolls in both hot and cold rolling. The average roll temperature in the cooling area during hot rolling starts above water boiling point and reduces close to room temperature. A temperature

of about 50°C is typical for cold rolling where emulsions are widely used.

Technical literature on cooling by emulsions and oils is not abundant. Several polymers were tested to determine the shearing effect of long molecules on physical properties and sprayed droplet size distributions through a flat fan nozzle [1]. Increasing concentration and density increased droplet sizes of spray samples. More recirculation cycles led to shorter molecular chains and then to smaller droplets, however, the droplets diameters were always at least slightly higher than those obtained with pure water.

Qiao and Chandra [2, 3] present their investigations on water with surfactant cooling. An additional surfactant in water decreases the parameters of surface tension, droplet diameter, and subsequently influences diameter and angle. A mass of surfactant also affects evaporation time, depending on surface temperature. Similar conclusions were found by Yang [4] for a wide range of surfactant types. Manzello and Yang have published a study, where they presented a liquid

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film thickness as a function of a water additive. Bertola and Sefine [5] described the behaviour of polymer additives impacting hot surfaces.

Despite the fact of oil cooling being widely used, there are not many published results on this topic. Wallis [6] presents optimized cooling for nickel super-alloys and titanium forgings. His study compares oil, water-air mist and fan water cooling through a range of 100 – 1100 °C. The oil provides a relatively lower heat transfer coefficient (HTC) of  $250 \text{ W.m}^{-2}.\text{K}^{-1}$  for temperature intervals of 100 – 400 °C. The air-mist and water-fan cooling show a different behaviour. High peaks can be observed at lower temperatures ( $4000 \text{ W.m}^{-2}.\text{K}^{-1}$  for 150°C) and lower HTC values in the high temperature region ( $200 - 300 \text{ W.m}^{-2}.\text{K}^{-1}$  above 500°C). According to Mudawar, Estes [7], Bolle and Moureau [8] the cooling intensity of oils can be enhanced by increasing their thermal conductivity, specific heat and density, which are lower than for water. The surface tension of oil is also lower and therefore the oil droplets are smaller. As mentioned in [8], bigger droplets lead to a higher cooling intensity in respect of the surface temperature range considered. The effect of additives in water is complex and there are no known methods on how its influence on cooling intensity can be theoretically quantified. The only possible approach is that provided by measurements.

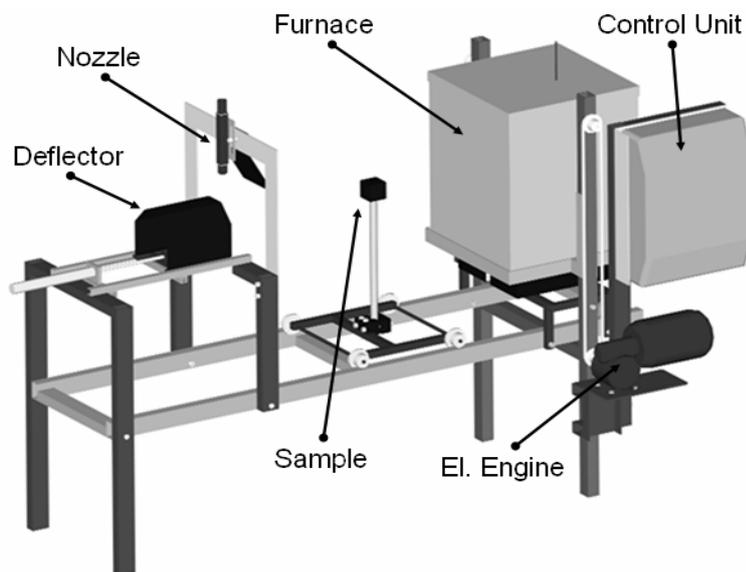
## Methodology

A plan of experiments was designed to quantify the influence of additives in water on cooling performance. The experiments were

performed on a test rig for spray cooling (**Figure 1**). The rig consists of an electric furnace in order to heat samples, a rig frame, a pool and a cooling section including the nozzle, a cooling circuit which allows diverse coolant utilisation and control unit. The nozzles used in all tests were a common industrial full cone nozzle (60° spray angle) with an impingement density of  $12.75 \text{ l.m}^{-2}.\text{s}^{-1}$  at 5 bars. A test sample from an austenitic steel plate ( $80 \times 80 \times 30 \text{ mm}$ ) is located 150 mm from the nozzle orifice. The sample is equipped with shielded thermocouples measuring temperature histories during the cooling in a depth 0.4 mm below the cooled surface.

The experiment starts by heating the sample to an initial temperature of 250°C. Then the sample is shifted to the cooling section, where it is protected from the coolant by a deflector unit. After that the deflector is opened and the cooling begins. During the experiments all data received from sensors are stored in a data logger for later inverse calculation and analysis.

Five types of coolants were chosen for the study (**Table 1**). The most widespread coolant is water, so the cooling intensity of other liquids is related to the cooling intensity of water. The emulsion 1 involves a mixture of water and mineral oil. Viscosity of the oil is similar to the viscosity of water. This emulsion is used for cooling rolls and lubricating the rolling gap in cold rolling. It was applied in a mass concentration of 2% and 4% mixture. The emulsion 2 (Lubrodal 192) is a mixture of water, polymers and organic salts.



**Figure 1.** Scheme of experimental

The experiments were done with concentrations varying from 0.1 % to 50%. In addition, the paper indicates the influence of water total hardness on the cooling process. In this respect, experiments were carried out with salt dilution,  $\text{CaCl}_2 \cdot 2\text{H}_2\text{O} + \text{MgSO}_4 \cdot 7\text{H}_2\text{O}$ .

**Results and Discussion**

Evaluation using inverse task

Measured temperature data from the sensors are used as input for inverse task calculating surface temperature and HTC as time-dependent values. Sequential estimation of the time varying boundary conditions algorithm is used for computation of thermal boundary conditions. Computational algorithm is based on minimizing differences between computed and measured temperatures in the sensor location. The method uses sequential estimation of the time varying boundary conditions and uses future time steps of measured data to stabilize the ill-posed inverse problem. To determine the unknown surface heat flux  $q$  at the current time  $m$ , the measured temperature responses  $T^*$  are compared with the computed temperature  $T$  from the forward solver using  $f$  future times steps

$$SSE = \sum_{i=m+1}^{m+f} (T_i^* - T_i)^2 \quad (\text{Eq. 1})$$

where  $SSE$  is sum of square errors. The used forward solver is based on finite volume method. Using the linear minimization theory, the value of the surface heat flux  $q$  that minimizes equation (1) is:

$$q^m = \frac{\sum_{i=m+1}^{m+f} (T_i^* - T_i|_{q^m=0}) \zeta_i}{\sum_{i=m+1}^{m+f} (\zeta_i)^2} \quad (\text{Eq. 2})$$

where  $T_i|_{q^m=0}$  are the temperatures at the temperature sensor locations computed from the forward solver using all the previously computed heat fluxes, but without the current one  $q^m$ . The  $\zeta_i$  is the sensitivity coefficient of the temperature sensor at time index  $i$  to the heat flux pulse at time  $m$ . These sensitivity coefficients physically represent the rise in temperature at the temperature sensor location for a unit heat flux at the surface. The sensitivity coefficient of our interest is defined as:

$$\zeta_i = \frac{\partial T_i}{\partial q_m} \quad (\text{Eq. 3})$$

The sensitivity coefficient is obtained from difference equation when two values of  $T_i$  are computed using a small change in boundary condition  $q_m$ .

Once the heat flux is found for the time  $m$ , the corresponding surface temperature  $T_m^{\text{surf}}$  may be computed using the forward solver. When the surface heat flux  $q_m$  and surface temperature  $T_m^{\text{surf}}$  are known the heat transfer coefficient is computed from:

$$HTC_m = \frac{q_m}{T_m^{\text{water}} - T_m^{\text{surf}}} \quad (\text{Eq. 4})$$

Once the HTC at the ‘‘present’’ time is computed, the time index  $m$  is incremented by one, and the procedure is repeated for the next time step. The basic approach is limited to linear problems. However, it can be extended to nonlinear cases. The modification of this procedure involves

**Table 1** List of used coolants and experimental parameters set up

Coolant type	Experiment parameters	Experiment name
Pure Water	pressure of 1, 3, 5, 7 and 10 bar	A1, A2, A3, A4, A5
Oil	pressure of 1, 3 and 5 bar	B1, B2, B3
Emulsion 1	concentration 2 and 4% in water; pressure of 5 bar	C1, C2
Emulsion 2	concentration 0.1, 1, 5, 10, 20 and 50 % in water; 5 bar	D1, D2, D3, D4, D5, D6
Salt Dilution	total hardness 16.55 and 30.1 mmol/L; pressure of 5 bar	E1, E2

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an outer iteration loop which continues until the computed temperature field is unchanging. The nonlinearity requires iteration only to determine the present value of the heat flux, while the computations to determine the surface temperature and heat transfer coefficient need only be performed once for each time  $m$ . The sensitivity coefficients are also nonlinear, due to the dependence of the thermal properties on the temperature field, and they must be computed for each iteration.

## Cooling Results Analyses

### Reference cooling tests with pure water

Spraying with pure water at a pressure of 5 bar provides a reference cooling intensity (exp. A3). The surface temperature history and computed HTC have been plotted in **Figure 2**. At the start of cooling, the surface temperature rapidly reduced from the initial temperature and at the same time the HTC abruptly increased. Cooling intensity is almost constant during the experiment. The dependence of HTC on water pressure can be seen in **Figure 3**. The average HTC values for surface temperatures in a range of 100°C to 30°C were computed for further evaluations. These values are shown in **Figure 4** together with the results of Hall and Mudawar [9] and Ciofalo [10, 11]. Both authors published results for low surface temperatures, water cooling by full cone nozzles and cooling of aluminium surfaces. The

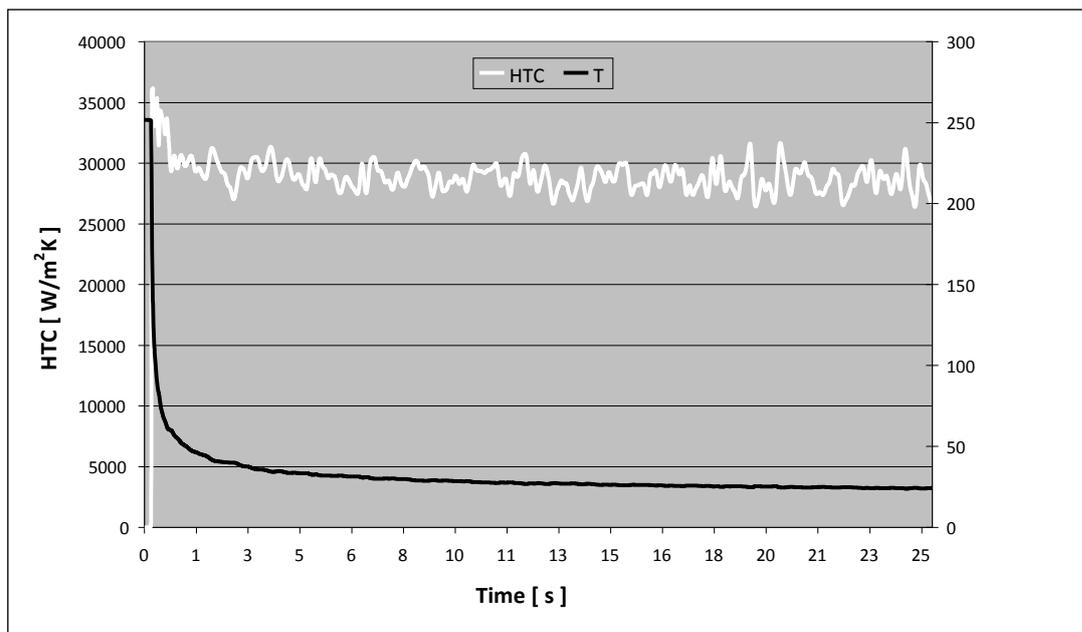
results for pure water on a steel surface obtained in this paper are between the results of Mudawar and Ciofalo and are used as a comparison reference for the cooling efficiency of oils and emulsions.

### Tests with oil

The experiments for oil were carried out at pressures of 1, 3 and 5 bar. It should be stressed that the viscosity and density of oil are different from the viscosity and density of water. Although the pressure was exactly the same in the experiments with water as with oil, the flow rate of oil was 15% higher through the nozzle due to viscosity difference. **Figure 4** shows a direct comparison of the cooling intensity of water and oil based on coolant impingement density. **Figure 5** provides an idea of the range of HTC due to pressure, coolant changes and their concentrations. The average HTC values during the spray cooling are in 78-81% lower for the tested oil than for water of identical liquid impingement density. The change of oil pressure results in a much lower impact than the change in water pressure but HTC behaviour is similar to pure water - such that an increase of pressure raises the average HTC significantly.

### Tests with emulsions

Two types of emulsions were tested. All of these tests were performed with a coolant pressure of 5 bar (**Table 1**). Emulsions and water provided identical flow rates for identical pressures.



**Figure 2.** Surface temperature and HTC history for pure water, pressure 5 bar, the reference experiment

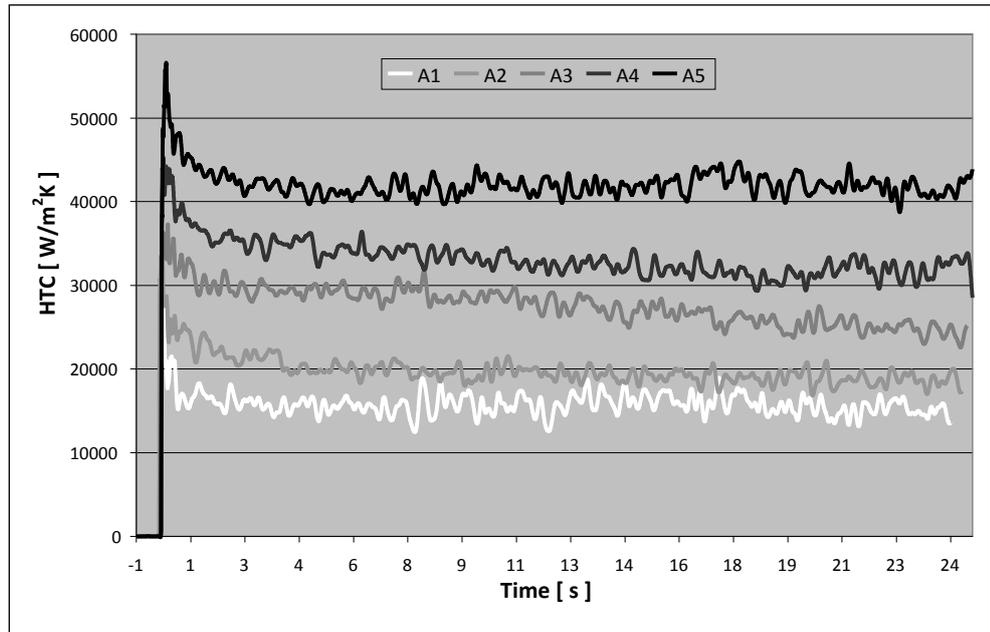


Figure 3. HTC history for pure water and growing pressure, 1, 3, 5, 7 and 10 bar

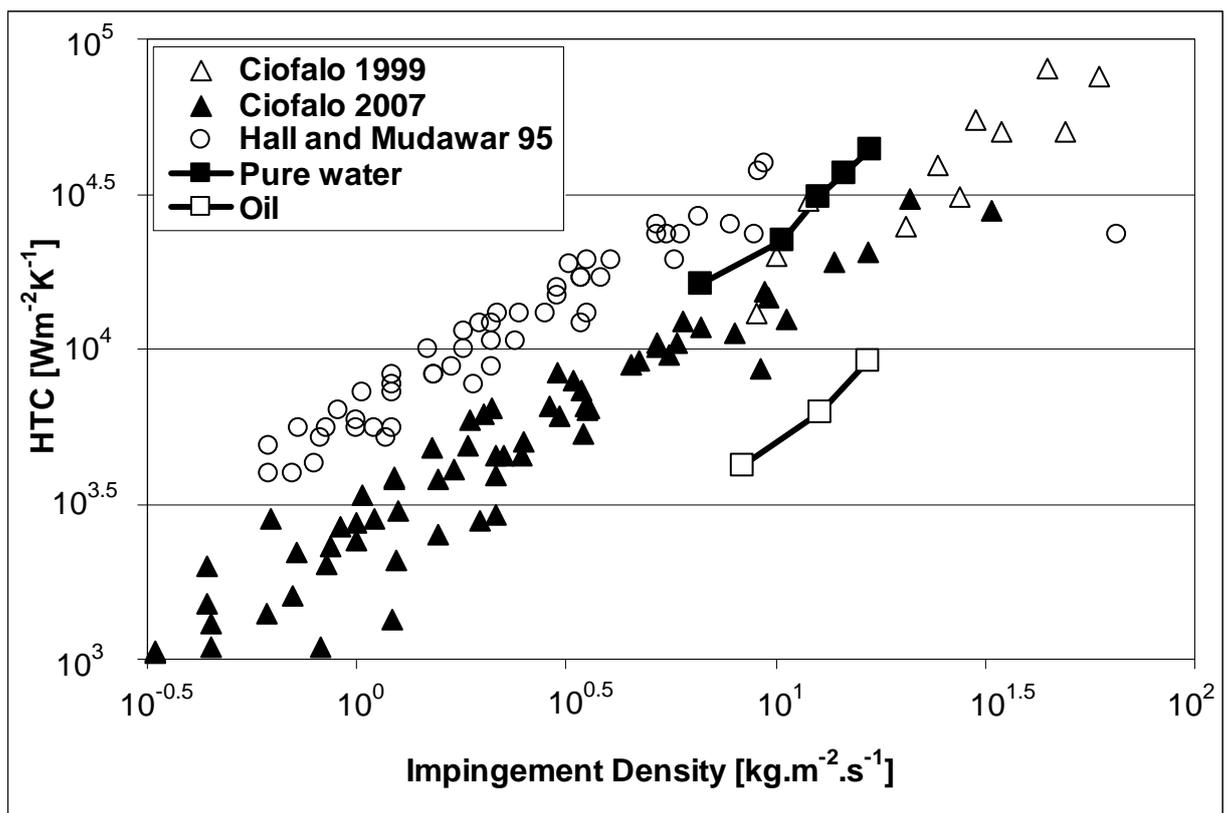
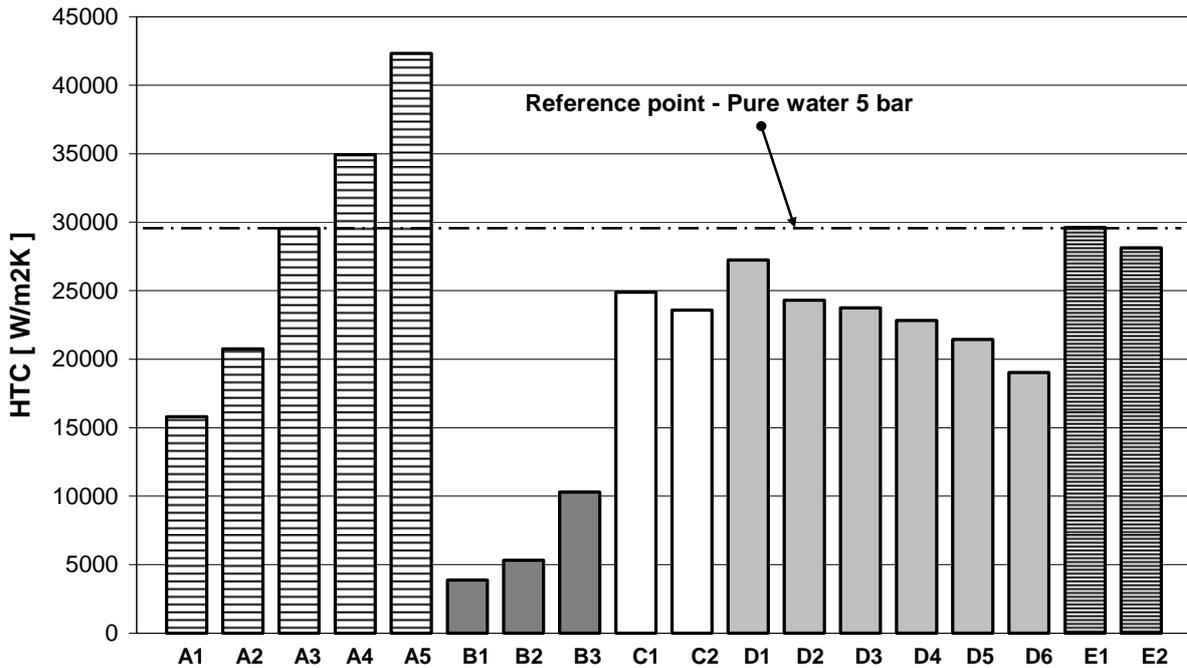


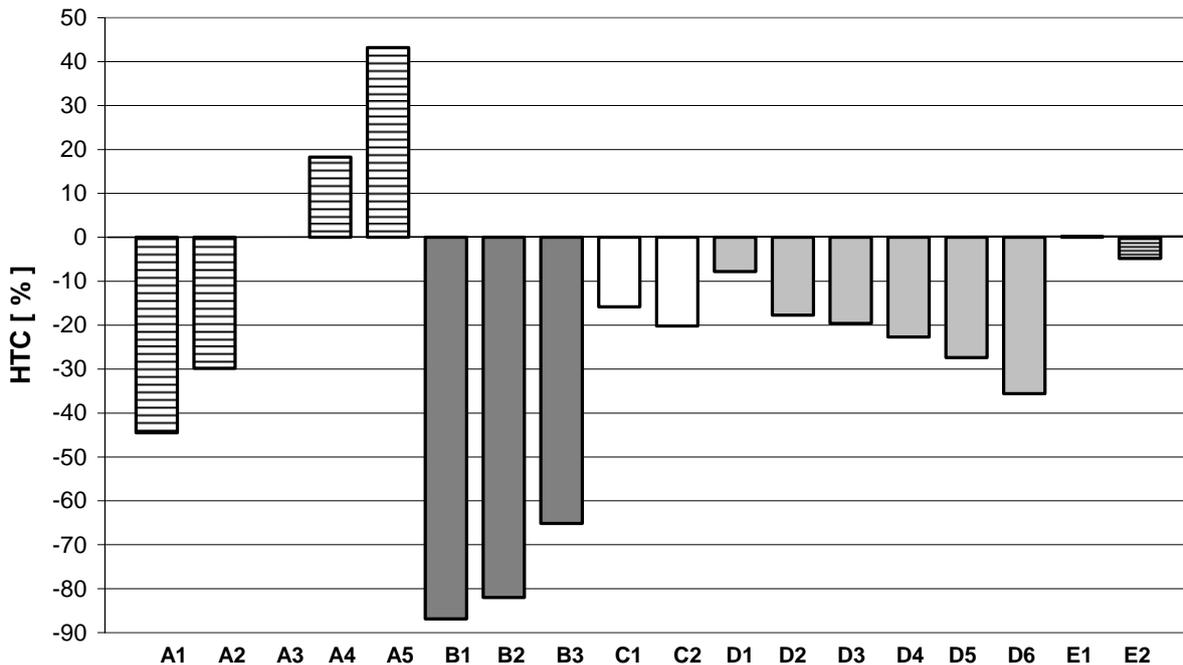
Figure 4. HTC values for water and oil

The changes of the cooling intensity are caused only by nature of liquids. The results obtained from this investigation are shown in **Figures 5 and 6**. Both emulsions show a decrease in average HTC value in comparison to the reference cooling

with water (exp. A3). The results show the HTC slumping trend for increasing concentration percentage. The emulsion 1 with a concentration of 2% and 4 % performs at only 84 and 79 % of water cooling performance under the same pressure



**Figure 5.** Comparison of average HTC values for all test cases



**Figure 6.** Comparison of relative HTC values, the reference point used is the cooling intensity of water at 5 bar (experiment A3)

conditions. Applying the emulsion 2 and higher admixture percentages results in a less intensive cooling. Using the following concentrations: 0.1, 1, 5, 10, 20 and 50 % caused the following HTC drops: 7.8%, 17.7%, 19.6%, 22.7%, 27.4% and 35.6 %. It is interesting to find that a main portion

of HTC declination comes from low concentrations. For example, the concentration change from 0 to 0.1 % caused a fall of 3 000  $W.m^{-2}.K^{-1}$  in absolute numbers, while the change from 20 to 50 % caused a fall of only 2 500  $W.m^{-2}.K^{-1}$ .

There are several factors which can explain the lower cooling efficiency of emulsions. Surface tension is reduced and the spray forms smaller droplets. Even a small percent of additives can increase the coolant viscosity and decrease the evaporation rate [1]. Higher viscosities reduce turbulent intensity in the coolant layer at the cooled surface. This, together with less evaporation, reduces heat transfer.

### Tests using high water hardness

The last batch of experiments covered the investigation of water salt dilution. It is obvious that even tap water contains a small number of dilutions. The level of salt contamination is related to water Total Hardness (TH). To find a relationship between TH and cooling efficiency,  $\text{CaCl}_2 \cdot 2\text{H}_2\text{O} + \text{MgSO}_4 \cdot 7\text{H}_2\text{O}$  admixture was introduced into the cooling water circuit. The cooling intensity was measured and compared to pure water (same reference point with a 5 bar pressure) using two dilutions with TH equal to 16.55 and 30.1 mmol/l. As shown in **Figures 5** and **6**, the difference between pure and diluted water is not significant. The variance is much lower than for changing pressure, coolant type and the concentration of coolants. Although TH 16.55 reaches 101 % (random statistical error) the salt additives are not expected to change the HTC values by a measurable level.

### **Conclusions**

A group of experiments using oil, polymers and salt diluted in water has been conducted. Spray cooling for conditions typical to that of cooling of rolls in hot and cold rolling was tested. These tests show that even very low concentration of additives in water can significantly reduce cooling intensity in comparison to pure water.

The following conclusions can be drawn:

- The most important finding is that water-based emulsions showed a measurable decrease in cooling intensity even for 0.1% of the oil content in water. The cooling intensity dropped by 7.8% for an oil content of 0.1% and by 17.7% for an oil content of 1%. These values are not negligible and should therefore be considered, particularly as changes in rolling technology can influence coolant quality. Low concentrations can be found in the polluted cooling water in industrial plants. A significant decrease in cooling efficiency should be taken into consideration when designing spray cooling systems and replacing water by another coolant.

- Spray cooling is significantly influenced by the coolant type used. Additives tested in water have shown a negative affect on the cooling process.

- Pure oils with a low viscosity used for the spray cooling, for example in aluminium rolling, have in 78-81% smaller cooling intensity than water with the equivalent impingement density.

- Water Total Hardness does not greatly influence the cooling intensity. The heat transfer coefficient decreases slightly (by a few percent), probably due to a surface layer of crystallized salts which forms during the evaporation process. A thick and non-homogenous layer is expected to be formed on the cooled surfaces.

- Oil admixtures and emulsions cause a significant drop in cooling intensity (as much as 40 %) compared to pure water under the same conditions.

- Surface sediments and burnt-on oil residues at the cooled surface create additional thermal resistance which can greatly reduce the intensity of heat transfer. Laboratory experiments showed that these thermal barriers formed during the spraying are of unpredictable thickness and can cause overheating of the cooled surface.

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