COOLING EFFICIENCY OF VARIOUS EMULSIONS DURING COLD ROLLING*

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Abstract

Emulsions are frequently used for the cooling and lubrication of working rolls during the cold rolling. The fact that the lubrication and cooling are not separable makes the optimization of both functionalities difficult. The higher emulsion temperature decreases the cooling efficiency compared to cold water. Heat transfer and fluid flow laboratory (Czech Republic), CRM (Belgium), Tata steel (Netherlands) and Henkel (Germany) cooperated on the increasing of a cooling efficiency during cold rolling. Part of this research was focused on an influence of the emulsion composition on a cooling intensity. The cooling efficiency was laboratory investigated. The stainless steel sample with thermocouple was heated and then it was cooled by a nozzle. Tests were performed with various emulsions and pure water. The heat transfer coefficient was computed from measured temperatures. The cooling intensity (heat transfer coefficient) of all tested emulsions was similar but an increase of the cooling intensity was found with water. Further various emulsions concentrations were tested. Results showed decreasing cooling intensity with increasing oil concentration. Adding additives therefore is thought to cause a noticeable decrease of the cooling intensity.

Keywords: Cold rolling cooling; emulsion concentration; cooling intensity, additives.

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1 INTRODUCTION

Mills continuously endeavor to increase the capabilities by increasing reduction per stand to roll harder material and at higher speeds. One of the main problem is the enormous heat generation in the roll bite which leads to oil-film breakdown and arising of scratches on the strip surface. One of the challenges is to develop an efficient cooling system that makes it possible to remove this heat from the roll without disturbing lubrication system. Emulsions are used in cold rolling process as a coolant and lubricating medium. Unfortunately the cooling intensity decreases with increasing oil concentration [1]. Literature studies [2],[3],[4],[5],[6],[7] showed possibilities of a cooling efficiency enhancing by adding different additives to the emulsion such as nano-particles [2] [3], polymers [4], surfactants [5] [6] and alcohols [7] [8]. Obtained conclusions from these articles are interesting. Higher oil viscosity leads to less efficient cooling (immersion cooling). Additives based on decreasing of contact angle could increase a heat transfer coefficient. Many surfactants leaded to an increase of the heat transfer coefficient.

Goal of this research was to investigate possibilities of enhancement of a cooling intensity. Six different coolants were chosen first: water and other three emulsions based on palm, coconut and lard oil. The last type of mentioned emulsion was used in cold rolling process, regenerated, and used during this study. Further various concentrations of surfactants and oils in emulsions were tested.

2 Experimental equipment, set up and procedure

A Static experimental stand was used for these tests. This stand was developed by Heat transfer and fluid flow laboratory (Figure 1). It is compound of electrical heater, thermal regulator, data-logger, manometer, thermal sensor, full cone nozzle, coolant temperature sensor, fluid pump, deflector, mixing and collecting chamber.

Figure 1. *Static experimental stand*

A thermal sensor with a diameter of 20 mm was built from austenitic stainless steel plate (Figure 2). Two shielded K-type thermocouples of diameter of 0.5 mm were inserted to the groves (made by electro-erosive machining) and soldered by silver.

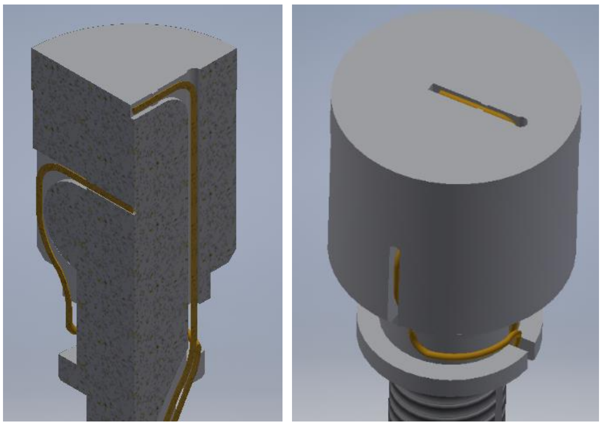

Figure 2. *Thermal sensor model (on the right), crossection of the thermal sensors with both thermocouples (on the left)*

Tomographic scan is shown in Figure 3. It is an example of the imperfect soldering. Cavities are visible under and around the thermocouple (sprayed surface is on the top of the sensor). So each used sensor have to be calibrated.

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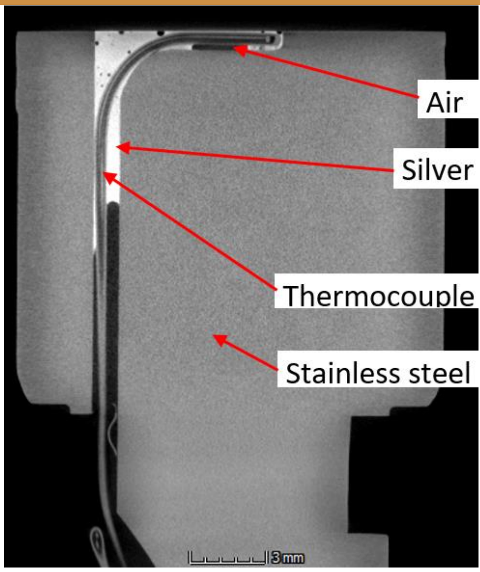

Figure 3. Tomographic scan of the imperfect temeprature sensor

Thermal sensor was fixed in Ketron-PEEK insulation cylinder (Figure 1, on the right bottom picture, temperature sensor in insulation inserted in stainless steel weight holder).

The electrical heater was composed of the cooper heating element which was inserted in insulated steel tube (Figure 1). The heater temperature was controlled by a regulator.

Lechler full cone nozzles 460.604 and 460.884 were used for these experiments. The nozzle was fixed in the center of the retaining chamber in the temperature sensor axis. A coolant temperature was measured by thermocouple which was inserted before the nozzle inlet.

Experimental procedure description follows. Water was heated to 51 °C first and then prepared for experiments (mixed with oil, with additives etc.). Mixing time of the coolant was longer than 30 minutes. Temperature sensor was cleaned and put on the heater (heating element) by tested surface down. Initial heating temperature was set to 250 C for all experiments and it was held more than 15 minutes to reach an uniform temperature field in the body. Afterwards a water pump was switched on and a pressure was set. Sensor was placed on the top of the collecting chamber cover in nozzle axis. Sensor sprayed surface was oriented down. Deflector was positioned between surface and nozzle. It was quickly removed when all of experimental parameters were set and the coolant was sprayed on the heated sensor surface until the final temperature was lower than 80°C. The fluid was collected in a chamber and recycled back to the tank. All thermocouples were connected with data-logger and measured temperatures were logged with frequency of 320 Hz. Time dependent boundary conditions (surface temperature, heat transfer coefficient and heat flux) were computed by inverse task and evaluated. Obtained data were compared in form of a heat transfer coefficient dependence on a surface temperature.

Inverse task calculation was used. This method is detailed described in [9]. Professor Pohanka developed and described software for computation of heat transfer coefficient, heat flux and surface temperature from measured temperature data under sprayed surface by 1D inverse task. The method uses Beck's sequential algorithm [10]. Estimation of the time varying boundary conditions uses future time step data to stabilize computation (0.01 s in our case). Determination of the unknown surface heat flux at time t^m is estimated by comparison of the measured temperature $\boldsymbol{T_r^{*,k}}$ with computed $\boldsymbol{T_s^k}$ from the forward solver (finite differential method) using n_t future time step

$$\boldsymbol{SSE} = \sum_{k=m+1}^{m+n_t} \sum_{s=r,r=1}^{n_T} \left(T_r^{*,k} - T_s^k\right)^2, \qquad (1)$$

where n_T is the number of thermocouples. Heat flux q^m in time step m. Using the linear minimization of (1), SSE → min, the value of the surface heat flux q^m is

$$q^m = q^{m-1} + \frac{\sum_{\kappa=m+1}^{m+n_t} \sum_{s\equiv r,r=1}^{n_T} \left(T_r^{*,\kappa} - T_s^{\kappa}\right) \cdot \zeta_r^{\kappa}}{\sum_{\kappa=m+1}^{m+n_t} \sum_{r=1}^{n_T} \left(\zeta_r^{\kappa}\right)^2}, \qquad (2)$$

where $\boldsymbol{T_s^k}$ are the computed temperatures by forward solver using all previously computed heat fluxes without the current one q^m. The ζ_r^κ is a sensitivity coefficient of the r[th] temperature sensor at time t^K to the heat flux pulse at time t^m. These sensitivity coefficients are partial derivatives of the computed temperature field to the heat flux pulse, but in this case they physically represent the rise in temperature at the temperature sensor location for unit heat flux at the surface. The sensitivity coefficient is defined as

$$\zeta_r^\kappa = \frac{\partial T_r^\kappa}{\partial q^\kappa}.$$

Once the heat flux is computed for a time t^m, the corresponding surface temperature and the heat transfer coefficient (h^m) is computed using following formula

$$\boldsymbol{h^m} = \frac{\boldsymbol{q^m}}{\boldsymbol{T^m_{coolant}} - \boldsymbol{T^m_{surface}}}$$

where $\boldsymbol{T^m_{coolant}}$ is measured during the experiment. The index m is incremented by one after heat transfer coefficient computation and the procedure is repeated for the next time step.[11]

3 RESULTS AND DISCUSSION

All of proceeded tests were done for four water flow rates (1.8 l/min – 17.5 l/min) with various water pressures (0.5, 2.5 and 5 bar). Two nozzles were used (460.604 and 460.884). Experimental combination specifying nozzle, water flow rate and water pressure combination is written in Table 1.

Table 1 Specification of used nozzles, coolant pressure and flow rate

Nozzle	Pressure [bar]	Flow rate [$lmin^{-1}$]
460.604	0.5	1.8
	5	4.5
460.884	0.5	9.2
	2.5	17.5

Typical result of these tests is shown in Figure 4. Heat transfer coefficient increased with increasing water flow rate and pressure except water flow rates of 9.2 $lmin^{-1}$ and 4.5 $lmin^{-1}$ which were comparable.

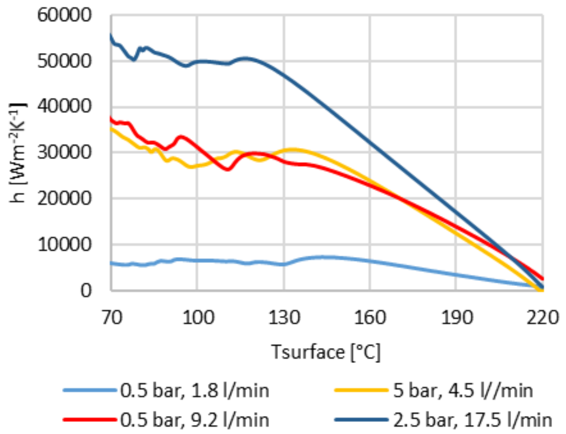

Figure 4. Comparison of different water flow rates and pressures for lard base oil

It showed strong dependence of the heat transfer coefficient on the coolant pressure and flow rate ratio. This effect was verified by simple computation below. Verification was done for water because droplet sizes were not measured but estimated from Lechler nozzle spray catalogue [12]. However similar behaving for stable emulsion was expected.

Nozzle producer measured droplet sizes for nozzles 460.604 and 460.964, dashed lines in Figure 5. But nozzles 460.604 and 460.884 were used. Droplet size values for corresponding water pressure but different nozzle were found by linear interpolation (Figure 5 full lines). Droplet sizes for investigated pressure were computed using formulas obtained by polynomial regression (Figure 5 red formula for nozzle 460.884 and green for nozzle 460.604).

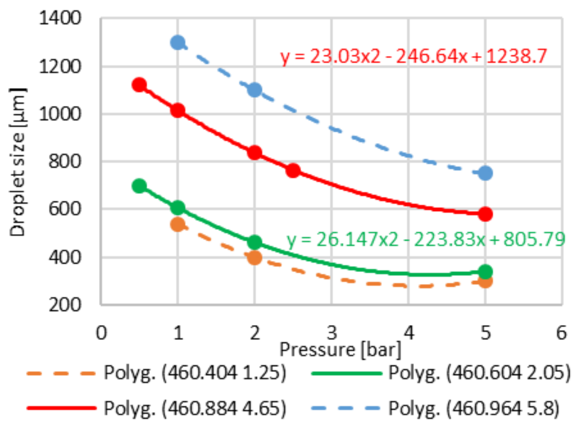

Figure 5. Computed droplet sizes (full lines) from nozzle producer catalogue values (dashed lines)

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Table 2. Computed droplet sizes for tested combinations of water flow rate and pressures

Nozzle	Bore diameter	Pressure	Flow rate	Droplet size
	[mm]	[bar]	[lmin^{-1}]	[µm]
460.604	2.05	0.5	1.81	700
460.604	2.05	5	4.54	340
460.884	4.65	0.5	9.19	1121
460.884	4.65	2.5	17.49	766

Following formulas were used to compute heat transfer coefficient. They were obtained in [13].

$$h = \frac{N_u \cdot \lambda}{L}, \qquad (3)$$

where h is a heat transfer coeficiient, Nu represents a Nusselt number, λ symbolize a thermal conductivity [Wm^{-1}K^{-1}] and L is a thickness [m].
Nusselt number is defined as:

$$Nu = 4.7 \cdot Re^{0.61} \cdot Pr^{0.32}, \qquad (4)$$

where Re is Reinolds number and Pr is Prandtl number. Water droplet size is one of Reynolds number parameter.

$$Re_{D_{32}} = \frac{\dot{m}_L \cdot D_{32}}{\mu}, \qquad (5)$$

where $\dot{m}_L$ is water impingement density [kg m^{-2}s^{-1}], D32 is Sauter droplet diameter and µ is dynamical viskozity [N s m^{-2}].

Prandtl number is defined as

$$Pr = \frac{c_p \cdot \mu}{\lambda}, \qquad (6)$$

where c_p is specific heat [j kg^{-1}K^{-1}], µ is dynamical viskozity and λ symbolize a thermal conductivity.

Figure 6 shows the comparison between computed and measured data. The effect of the similar cooling intensities for two varisou flow rates is clearly visible in measured and predicited heat transfer coefficient.

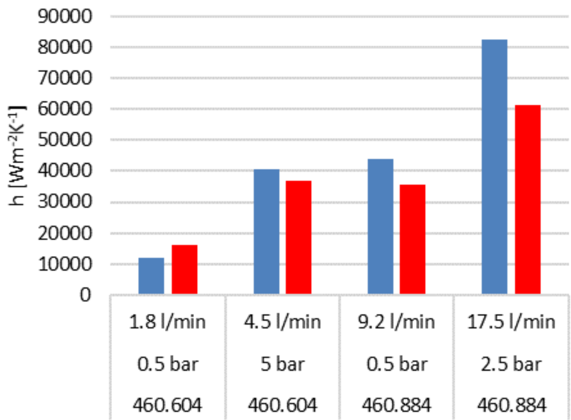

Figure 6. Comparison of the heat transfer coefficient for tested combinations of water flow rate and pressure, the blue lines represent measured data (averaged in surface temperature interval between 70 – 90 °C) and the red lines represent predicted heat transfer coefficient using Nusselt number

Afterwards the cooling process repeatability was investigated for all prepared emulsions. Example of achieved repeatability is shown in Figure 7. Repeatability of the experimentComparable repeatability was found for all tested emulsions.

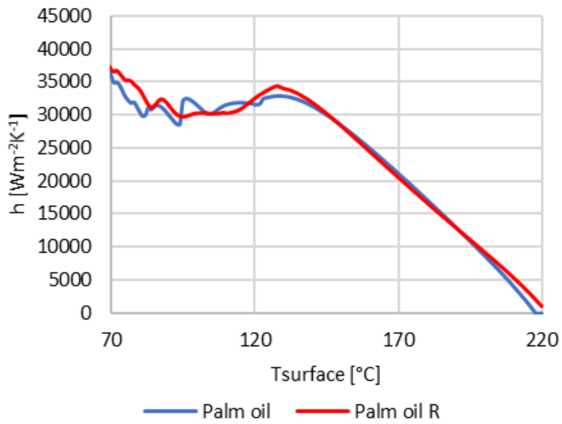

Figure 7. Repeatability of the experiment

Five various coolants were compared. Four emulsions and pure water. Emulsions were made by mixing water with 1% of: palm oil, coconut oil, lard oil and regenerated lard oil (after cold rolling). Figure 8. Heat transfer coefficient dependence on the surface temperaure for emulsions based on various oils and for pure water. Interesting results were found such as obtained heat transfer coefficient was similar for all emulsions, independent on base type. The heat transfer coefficient was higher only for water.

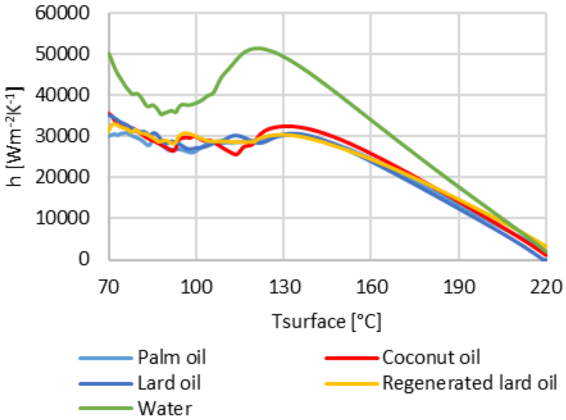

Figure 8. Heat transfer coefficient dependence on the surface temperaure for emulsions based on various oils and for pure water

Regenerated oil could be low cost alternative to “fresh” oil. The disadvantage of the regenerated oil is its dirtiness. Figure 9 shows thermal sensor after tests with regenerated oil. Part of the surface was cleaned to distinguish the difference. Cleaning the experimental equipment took longer comparing with other emulsions.

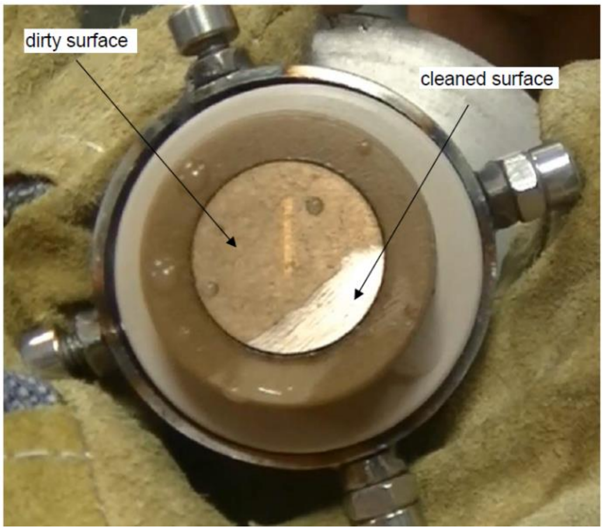

Figure 9. Sensor surface after the test with regenerated oil

Presence of 1% oil concentration in water decreased significantly a heat transfer coefficient. The next step was to test lower concentrations, 0.1, 0.3 and 0.6%. Results are shown in Figure 10. Interesting results was that the heat transfer coefficient was similar for 0.1 and 0.3% as well as 0.6 and 1%. Even small concentration of oil (0.1%) created oil film on the temperature sensor surface and decreased the heat transfer coefficient Figure 10.

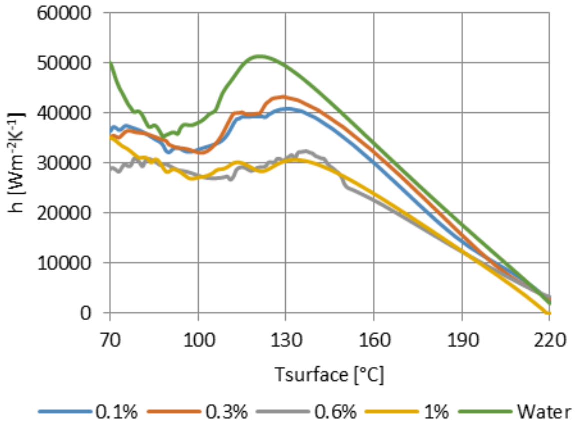

Figure 10. Comparison of various oil concentrations in emulsion, lard based oil

Next idea was to use surfactants. Pure water and mixture of water and surfactants were compared first. Used surfactant was made by Henkel, called Bonderite L-AD EP 5501. It was just one of the surfactants currently used in Henkel’s rolling oil formulations to adjust particle size and stability in emulsion. Two various concentrations (0.1 and 0.3%) were tested and compared with water. Presence of surfactant decreased heat transfer coefficient in the surface temperature area between 70 and 130°C. (Figure 11).

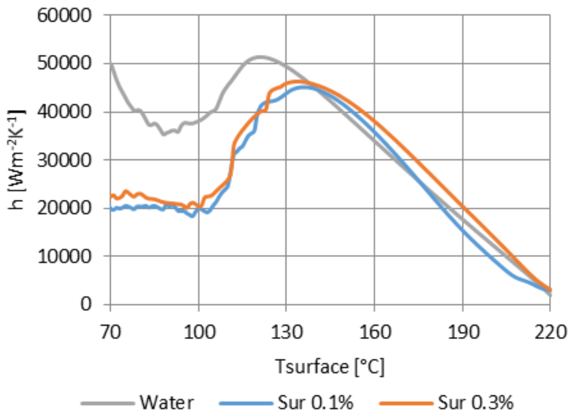

Figure 11. Comparison between cooling using pure water and two different surfactant concentrations

Finaly various mixtures of oil and surfactant in water were compared. Figure 12 shows the results of tests with constant concentration of surfactant (0.3%) and various percentage of oil (0 – 0.6%). The heat transfer coefficient decreased with increasing oil concentration.

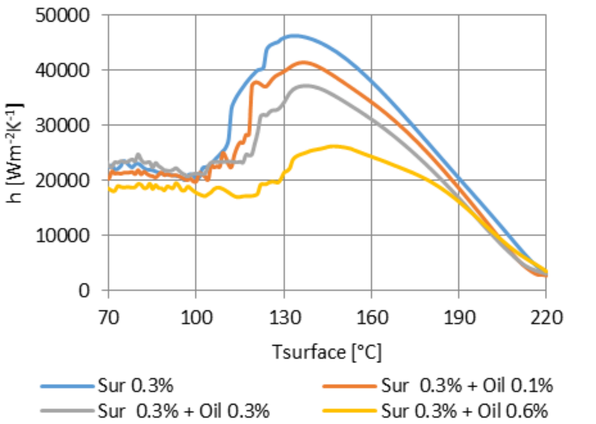

Figure 12. Comparison for different oil and surfactant concentrations in water, lard based oil

Comparison of changing surfactant concentrations in emulsion showed that the presence of surfactant decreased the heat transfer coefficient but together with increasing concentration of surfactant the heat transfer coefficient increased. This result was observed for both oil constant concentrations of 0.1 and 0.3% (Figure 13 and Figure 14).

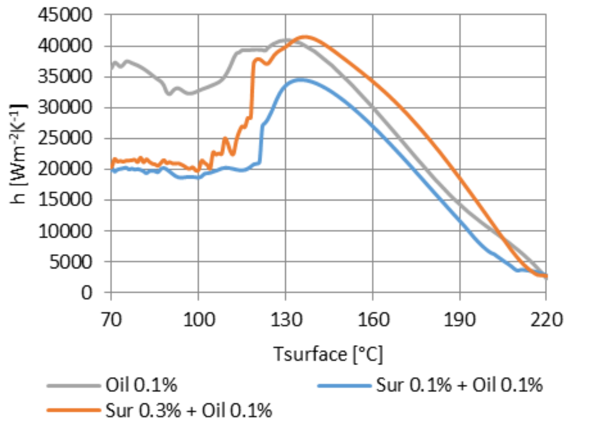

Figure 13. Comparison for constant oil and different surfactant concentrations in water, lard based oil 0.1%

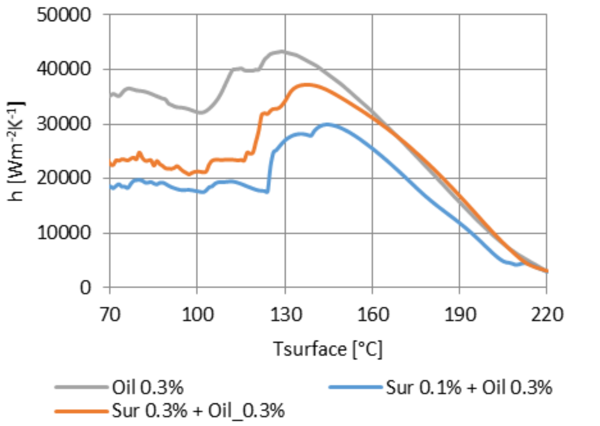

Figure 14 Comparison for constant oil and various surfactant concentrations in water, lard based oil 0.3%

4 CONCLUSION

The goal of this study was to find possibilities to enhance cooling intensity during cold rolling. Various emulsions were tested. These emulsions were based on palm, coconut and lard oil. All of four tested emulsions were cooled with the same intensity. Even similar heat transfer coefficient was found for regenerated (after rolling) lard based emulsion as for a fresh emulsion. Presence of oil in water decreased a heat transfer coefficient compared to pure water. Further, various oil concentrations were tested. Even small amount of oil in water (0.1%) caused a decrease of a heat transfer coefficient. Next logical step was to try to add surfactant to the emulsion which helps with emulsion stability. Surfactant presence in water caused decrease of a heat transfer coefficient comparing with pure water. Afterwards the various surfactant concentration in emulsion was tested. Very interesting results were achieved. Higher surfactant concentration increased the heat transfer coefficient but it was still lower than for emulsion cooling without surfactant.

Acknowledgments

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REFERENCES

1 Horak A, Raudensky, M, Pohanka M, Bellerova H, Reichardt T, Research on cooling efficiencies of water, emulsions, and oil. Journal of Metallurgical and mining industry. 2010; 2: 271-278
2 Bellerova H, Pohanka M, Spray cooling by multi-walled carbon nanotubes and Fe nanoparticles. WIT Transactions Modelling and Simulation, 2011, 51: 293-304
3 Ravikumar V, Jha M, Haldar K, Pal K, Chakraborty S, Surfactant-based Cu-water nanofluid spray for heat transfer enhancement of high temperature steel surface. Journal of heat transfer. 2015; 137: 1-8
4 Sarkar I, Behera K, Jha M, Pal K, Chakraborty S, Effect of polymer additive on the cooling rate of a hot steel plate by using water jet. Experimental thermal and fluid science. 2016; 70: 105-114
5 Ravikumar V, Jha M, Sarkar I, Mohapatra S., Pal K, Chakraborty S, Achievement of ultrafast cooling rate in a hot steel plate by air-atomized spray with different surfactant additives. Experimental thermal and fluid science. 2013; 50: 79-89
6 Ravikumar V, Jha M, Sarkar I., Pal K, Chakraborty S., Mixed-surfactant additives for enhancement of air-atomized spray cooling of a hot steel plate. Experimental thermal and fluid science. 2014; 55: 210-220
7 Ravikumar V, Jha M, Sarkar I., Pal K, Chakraborty S., Enhancement of heat transfer rate in air-atomized spray cooling of a hot steel plate by using an aqueous solution of non-ionic surfactant and ethanol. Applied thermal engineering. 2014; 64: 64-75
8 Cheng W, Xie B, Han F, Chen H, An experimental investigation of heat transfer enhancement by addition of high-alcohol surfactant (HAS) and dissolving salt additive (DSA) in spray cooling. Experimental thermal and fluid Science. 2013; 45: 198-202
9 Pohanka, M, Technical experiment based inverse tasks in mechanics. Ph.D. thesis. Faculty of mechanical engineering, Brno University of technology, Czech republic, 2006
10 Beck V, Blackwell B, Charles C, Inverse heat conduction, Ill-posed problems. Wiley, New York, 1985. ISBN 0–471–08319–4
11 Pohanka M, Bellerova H, Raudensky M, Experimental technique for heat transfer measurements on fast moving sprayed surface. Journal of ASTM international, 2009, 6: 3-15
12 Lechler, Main catalogue General Industry | Precision Nozzles and Accessories, 2012, 112:7, Avaliable from: https://www.lechler.com/fileadmin/media/kataloge/pdfs/industrie/katalog/EN/lechler_catalogue_112_en.pdf
13 RIBICKY R; MUDAWAR I, Single-phase and two-phase cooling characteristics of upward-facing and downward-facing sprays. International Journal of Heat and Mass Transfer, 2006, 49: 5–16.