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Numerical Simulation of Ice Slurry Flow in improved Plate Heat Exchanger Geometries with Consideration of different Phase Interaction Parameters

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ABSTRACT

Concerning the cold supply, ice slurry technology is a safe, environmentally friendly and efficient solution for energy storage. Previous studies have shown that ice slurry is potentially one of the most important phase change material (PCM) slurries used as a secondary refrigerant due to its high cooling capacity and flexibility in application. However, agglomeration phenomena are observed with ice particle slurry flows, which can lead to blocking of e.g. heat exchangers and pipes. The study applies an Euler-Euler approach based on the kinetic theory of granular flow to describe the melting of an ice slurry flow. The Eulerian model was used to simulate the ice slurry in new channel structures of specially manufactured plate heat exchangers. The mass flow rate, ice concentration, and industrial ice particle sizes were varied. The validation with experimental data is presented. The results show the pressure drop and agglomeration reduction with the new channel design relative to commercially available geometries of the plate heat exchanger. The simulation results show the velocity profiles, the volume fraction, and the pressure loss related to the ice concentration. The simulation results of improved heat exchanger plates to avoid high volume fractions at the inlet and outlet as well as the channels of specially designed plate heat exchanger are displayed. The results show the effect of changing the phase interaction parameter at high velocity.

Keywords: Ice Slurry, Euler-Euler Approach, Agglomeration, Plate Heat Exchanger, Channel Structures, Simulation

1. INTRODUCTION

Plate heat exchangers generally cost less than shell and tube heat exchangers because they are easier to manufacture. In addition, less material is required for the same transferred heat flow. At the same time, the plate design means that there are considerably more flow disturbances and promotes blocking by solids. A pipe loop construction is less susceptible to this due to the curves used. Therefore, the most robust plate heat exchanger with the lowest pressure drop and the best heat transfer properties has to be found. There are few experimental studies of ice slurry flows inside of plate heat exchangers; it was found that high pressure drops, i.e. agglomerations and blockages, occur in channels of plate heat exchangers. J. Bellas et al. (J. Bellas 2002) presented several experimental results specific to 5% propylene-water slurries melting in a commercial plate heat exchanger, where the mean ice fraction is between 5% and 20%. Their investigations show that the pressure drop increases with increasing flow rate and ice fraction, and that it affects the increasing overall heat exchanger heat transfer coefficient with increasing flow rate but after short time, the blocking and high pressure are happened.. G.S.F. Shire et al. (Shire, Quarini et al. 2009) presented the pressure drop of an ice slurry in the most complex process equipment (plate heat exchangers). Blocking was observed for high ice fractions or for large ice crystals (or agglomerations of smaller crystals). Fernandez-Seara et al. (Fernández-Seara and Diz 2014) investigated the pressure drop and heat transfer characteristics of an offset strip-fin plate heat exchanger operating with an ice slurry. Their experiments showed that instabilities in the flow rate occur under isothermal conditions due to complete or partial blockage of some of the heat exchanger channels for the lowest ice slurry flow rate. Additionally, a slight increase of heat transfer rate and the overall heat transfer coefficient with increasing ice fractions was observed. The numerical simulation of different structures of plate heat exchangers has already been carried out in Hefny et al. (Shaimaa HEFNY 2020), but it has not been validated. This is because the simulation of a complete plate heat exchanger does not make sense due to the high computing effort. Experimental results of complete plate heat exchangers geometries are given in the previous literature, which clearly show the problem of blocking by ice particles. In order to be able to make a prediction for blockages, various approaches for individual plate heat exchanger channels were investigated within the scope of a research project funded by the German Federal Ministry for Economic Affairs and Energy (BMWi, funding code: 03ET1476B) in this paper. In this part of research, the simulation of an ice slurry was performed for a complete Plate Heat Exchanger (PHE) geometry of THERMOTECHNIK ZEESEN GmbH & Co.KG (TTZ GmbH). These cases were validated with experimental data of pressure drops resulting from measurements at the Institute of Air Handling and Refrigeration (ILK), Department for Applied Energy Engineering in Dresden, Germany.

2. CFD MODEL

The simulation of the ice slurry was performed for a complete channel of the special plate heat exchanger from the project partner TTZ Company. The simulations included inlet ice volume fractions of 0% (liquid only) and 20% for a liquid-particle mixture of water with the potassium format solution. Three fluid zones are present: one fluid zone of ice slurry is located between two fluid zones of water, as shown in Figure 1. The various values of flow rates were 1.86 – 3 l/min and 5 l / min. The temperature of the water flow in two liquid zones is about 12℃.

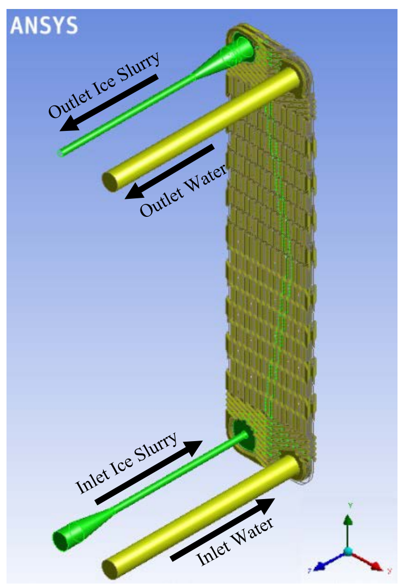

Figure 1: Schematic geometry of special plate heat exchanger for CFD model

A three-dimensional Euler-Euler granular multiphase turbulence model based on the kinetic theory of granular flow was used for the simulations. In the Euler-Euler approach, ice particles and additive solution mixing with water were treated as continua. An option used in solving this problem starts with a simpler mixture multiphase model and then choses the Eulerian model. It considers the effects of phase interactions, which include the interphase forces of drag, lift, turbulence dispersion, particle-particle interaction, and particle-wall collisions by incorporating interphase terms and the turbulence model. The mathematical model was presented in the independent publications in Hefny et al. (2020a) for simulating an ice slurry inside tube flow. The governing equations for each phase and the solver method are included. After the simulation of the ice slurry inside plate heat exchanger, it was found the interaction parameters (forces and shear stress viscosities) and the turbulence model must be modified to achieve convergence and good validation with the experimental results.

2.1 The Interaction Parameters

The interaction parameters play an important role during simulation through effecting the behavior of ice particles inside the liquid flow. The forces acting on particles through the channel of plate heat exchanger were analyzed as shown in Figure 2.

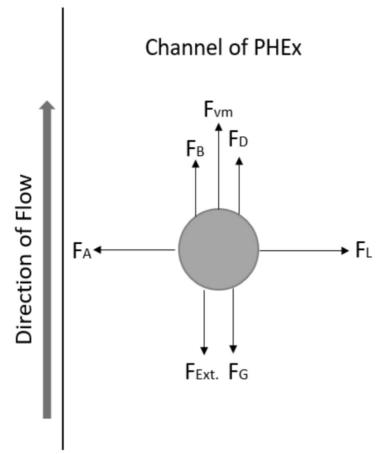

Figure 2: Forces acting on a particle through the channel of plate heat exchanger

The virtual mass force ($\overrightarrow{F_{vm}}$) presents itself when the secondary solid particle phase accelerates relative to the primary liquid phase. It is possible to include the virtual mass force ($\overrightarrow{F_{vm}}$) that is present when the secondary solid particle phase accelerates relative to the primary liquid phase for both granular and non-granular flows. The virtual mass effect is significant when the secondary phase density (ice density) is much smaller than the primary phase density (liquid density); the virtual mass force is defined as:

$$\overrightarrow{F_{vm}} = C_{vm}\alpha_s\rho_l\left(\frac{d_l\overrightarrow{U_l}}{dt} - \frac{d_s\overrightarrow{U_s}}{dt}\right) \tag{1}$$

Where C_{vm} is the virtual mass coefficient, which typically has a value of 0.5 in reference (D. A. Drew 1993) that included the studies of the coefficient C_M is equal to 0.5 because the bubble/particle is supposed to be spherical. Indeed, this coefficient only depends on the shape of the particle. In addition, the particle assumed in this work as a spherical.

The Drag Force ($\overrightarrow{F_D}$): is calculated by using the drag force coefficient according to Schiller-Naumann model (Schiller 1935):

$$\overrightarrow{F_{D,sl}} = k_{sl}\left(\overrightarrow{U_s} - \overrightarrow{U_l}\right) \quad , \quad \overrightarrow{F_{D,ls}} = k_{ls}\left(\overrightarrow{U_l} - \overrightarrow{U_s}\right) \tag{2}$$

It can be seen in the momentum equations that the momentum exchange between the phases is based on the fluid-solid and solid-solid coefficients K_{sl} for granular flows. The Gidaspow model (GIDASPOW 1994, Guide 2018) is a combination of the Wen and Yu model and the Ergun equation, and the fluid-solid exchange coefficient is of the following form:

For $\alpha_l > 0.8$, $\alpha_s \leq 0.2$

$$k_{sl} = \frac{3}{4}C_D\frac{\alpha_s\alpha_l\rho_l|\overrightarrow{U_s} - \overrightarrow{U_l}\,|}{d_s}\alpha_l^{-2.65} \tag{3}$$

The drag coefficient is modelled using an empirical correlation which was developed for laminar flow by (Schiller 1935):

$$C_D = \begin{cases} \frac{24(1+0.15{R_e}^{0.687})}{Re} & if\ Re \leq 1000 \\ 0.44 & if\ Re > 1000 \end{cases} \tag{4}$$

The Lift Force ($\overrightarrow{F_L}$): is the force perpendicular to the direction of flow (the drag force) due to the shear stress of fluid. For multiphase flows, the lift forces are included on the secondary phase particles. The lift forces act on the particle mainly due to the velocity gradients in the primary phase liquid flow field. The lift force will be more significant for larger particles. From Drew (D. A. Drew 1993), the lift force acting on a secondary phase of solid particle(s) in a primary phase of liquid can be calculated as:

$$\overrightarrow{F_L} = C_L \alpha_s \rho_l \left(|\overrightarrow{\mathrm{U}_s} - \overrightarrow{\mathrm{U}_l}| \right) . (\nabla . \overrightarrow{\mathrm{U}_l}) \tag{5}$$

The model developed by Moraga et al. (F.J. Moraga 1999) is applicable mainly to the lift force on spherical solid particles, though it can be applied to liquid drops and bubbles. In this model the lift coefficient combines opposing actions of two phenomena which are the interaction between the dispersed phase particles and the primary phase shear the vorticity-induced lift resulting from interaction between particles and vortices shed by particle wakes. the Moraga lift coefficient is formulated as:

$$C_L = \begin{cases} 0.0767 & \varphi \leq 6000 \\ -(0.12 - 0.2e^{-\frac{\varphi}{3.6}\times 10^{-7}}) & 6000 < \varphi < 5 \times 10^7 \\ -0.6353 & \varphi \geq 5 \times 10^7 \end{cases} \tag{6}$$

$$\varphi = Re_p Re_\omega \tag{7}$$

Where Re_p: particle Reynolds number and Re_ω:vorticity Reynolds number

The Buoyancy Force (F_B): is the force acting opposite the direction of gravity that affects all objects submerged in a fluid. When an object is placed in a fluid, the weight of the object pushes down on the fluid (liquid or gas) while an upward buoyancy force pushes upward on the object, acting against gravity.

The Gravity Force (F_g): is the force due to the gravity and acts in the opposite direction of the buoyance force.

The External Force ($F_{Ext.}$): is the force due to the pressure gradients of flow.

The Attraction Force (F_A): is the force due to the particle-wall interaction; in the case of pipes, the attraction can be neglected due to the dimension of the pipe compared to the dimension of the channel of plate heat exchanger.

3.1.8 The coefficient of turbulent dispersion force is defined by the Burns model (Alan D. Burns 2004). For multiphase turbulent flows, the Eulerian model can include the effects of turbulent dispersion forces, which account for the interphase turbulent momentum transfer. The turbulent dispersion force acts as a turbulence diffusion in dispersed flows. For the Burns model (Alan D. Burns 2004), it is estimated by the turbulent viscosity of the continuous phase:

$$\overrightarrow{F_{TD,s}} = \overrightarrow{F_{TD,l}} = C_{TD} k_{sl} \frac{D_s}{\sigma_{ls}} \left(\frac{\nabla \alpha_s}{\alpha_s} - \frac{\nabla \alpha_l}{\alpha_l} \right) \tag{8}$$

where $D_s = D_l = D_{t,s} = \frac{\mu_{t,s}}{\rho_s}$, $C_{TD} = 1$, $\sigma_{ls} = 0.9$, and $\mu_{t,s}$ is turbulent viscosity of the solid phase:

$$\mu_{t,s} = \rho_s C_\mu \frac{k^2}{\varepsilon} \tag{9}$$

The interface area concentration is defined by the ia-particle method (Guide 2018). Interfacial area concentration is defined as the interfacial area between two phases per unit mixture volume. This is an important parameter for predicting mass, momentum and energy transfer through the interface between the phases. The algebraic interfacial area models are derived from the surface area to volume ratio, defined for a spherical particle as:

$$A_s = \frac{\pi d_s^2}{\frac{1}{6}\pi d_s^3} = \frac{6}{d_s} \tag{10}$$

Particle Model: For a secondary phase (solid ice particle phase) volume fraction, α_s, the particle model estimates the interfacial area concentration Ai as:

$$A_i = \alpha_s A_s = \frac{6\alpha_s}{d_s} \tag{11}$$

2.2 The Turbulence Model

The different turbulence models were tested to obtain usable results and the corresponding convergence behavior of the models in different structures of the heat transfer geometries of the heat exchangers (tubes and PHE). It was found that the RNG (called renormalization group theory) of the k-ε turbulence model for ice slurry within the pipe and the realizable k-ε turbulence model for the ice slurry within the PHE are appropriate. There is a model for dispersed and the model for each phase, which gives good results when testing different ice particle diameters and ice agglomerate sizes with convergence calculations. The dispersed turbulence model is the appropriate model when the concentrations of the secondary phases are dilute.

3. NUMERICAL SIMULATION

In this part of research, the simulation of the ice slurry was performed for a special PHE geometry from TTZ Company with an inlet ice volume fraction of 0% (liquid only) and (5%-20%) for a liquid-particle mixture. These cases were validated with experimental data of pressure drops resulting from measurements at the ILK Dresden.

3.1 Single Phase (Only Liquid)

The comparison between the simulation and two experimental measurements from ILK Dresden for single-phase volume flow ranging from 1 l/min to 6 l/min is shown in Figure 3. Due to the parallel plates in PHEs, it was reasonable to assume that the pressure drops are almost equal. However, high pressure losses in the distribution at the inlet and outlet are apparent.

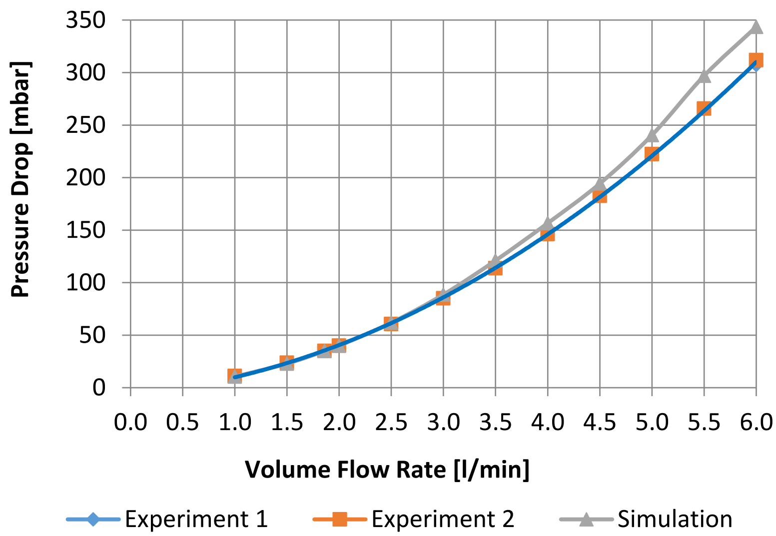

Figure 3: Pressure losses of the single-phase potassium format-water flow as a function of the inlet volume flow in the special plate heat exchanger geometry of TTZ Company

3.2 Two Phases (Liquid and Ice Particles)

The simulation of the ice slurry was carried out for a complete PHE geometry from TTZ GmbH with an inlet ice volume fraction of 0% (liquid only) and 20% for a liquid-particle mixture. These cases were validated with the experimental data of the pressure drops resulting from the measurements of the ILK Dresden. The simulation results agree very well with the experimental results at low flow rates (e.g. 1.86 l/min or 3 l/min), as shown in Figure 4. A deviation between the simulation and experimental result of more than 15% can be observed at 5 l/min. The reason for this is assumed to be the change in the interaction parameters of the moving particles, such as collision and frictional viscosity of the secondary fluid (ice particles). Furthermore, a frictional component of the viscosity can be included to account for the visco-plastic transition that occurs when particles of a solid phase reach the maximum solid volume fraction at high velocity (i.e. at high volume flow). It has been found that the models used for low flow velocities and low ice volume fractions do not provide satisfactory results at higher flow rates and concentrations. The Schaeffer model (Schaeffer 1987) was identified as particularly suitable for higher flow rate cases in order to model the behavior of the flow. It was found that under the updated model (dotted line in Figure 4) the numerical results show improved agreement with the experimental results.

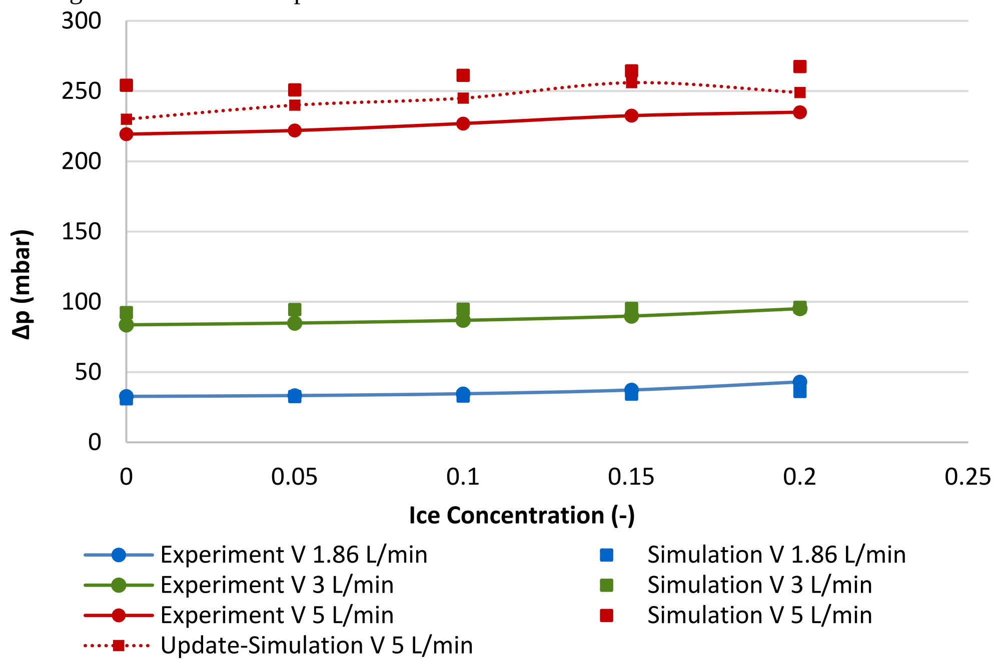

Figure 4: Validation of the pressure drop between experimental and simulation results of the special plate heat exchanger geometry of TTZ Company

However, it is noticeable that high-pressure losses exist mainly in the areas of the inlet and outlet as shown in Figure 5. Therefore, the simulation result of the investigation shows that the positions of the inlet and outlet openings on the sides (bottom and top respectively) of regular plate heat exchangers are not practicable for ice slurry flows.

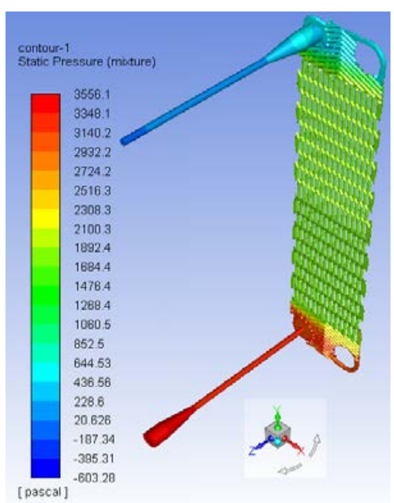

Figure 5: Static pressure of the mixture flow at V=1.86 l/min and ice volume fraction 20% inside the special plate heat exchanger geometry of TTZ Company

At the ILK Dresden a test stand for the visualization of ice slurry flows with the help of transparent plates was carried out. With the help of appropriate lighting and camera technology, images of agglomerations and blockages by the ice particle flow were taken and made available for comparison with the simulation profiles. The simulations were carried out for unsteady slurry ice flow over the course of 90 minutes in the previously described special design of the TTZ Company. Figure 6 shows the comparison of the flow visualization after 90 minutes of operating time with the simulation results for the inlet of ice slurry flow at ice volume fraction 20% and the volume flow rate 1.86 l/min. A bright colour in the two photos means that there is a constant flow in the channels. A dark colour means that there is a high proportion of ice and is an indication of agglomeration or blockage at the inlet of plate heat exchanger (bottom) in a comparison with the ice volume fraction profile which has a high volume fraction around 0.20 at the inlet (green colour). The regular inlet and outlet at the sides of the special plate heat exchanger are not practical for ice slurry flows. The ice particles elect for the shortest way from the inlet to the outlet opening and agglomerate in these channels, while channels with longer distances between the inlet and outlet remain almost unaffected. In addition, the velocity profile of ice particles shows the moving distribution of particles inside the channels. The dark blue colour is a very low velocity (approximately zero) and indicates agglomeration or ice melting.

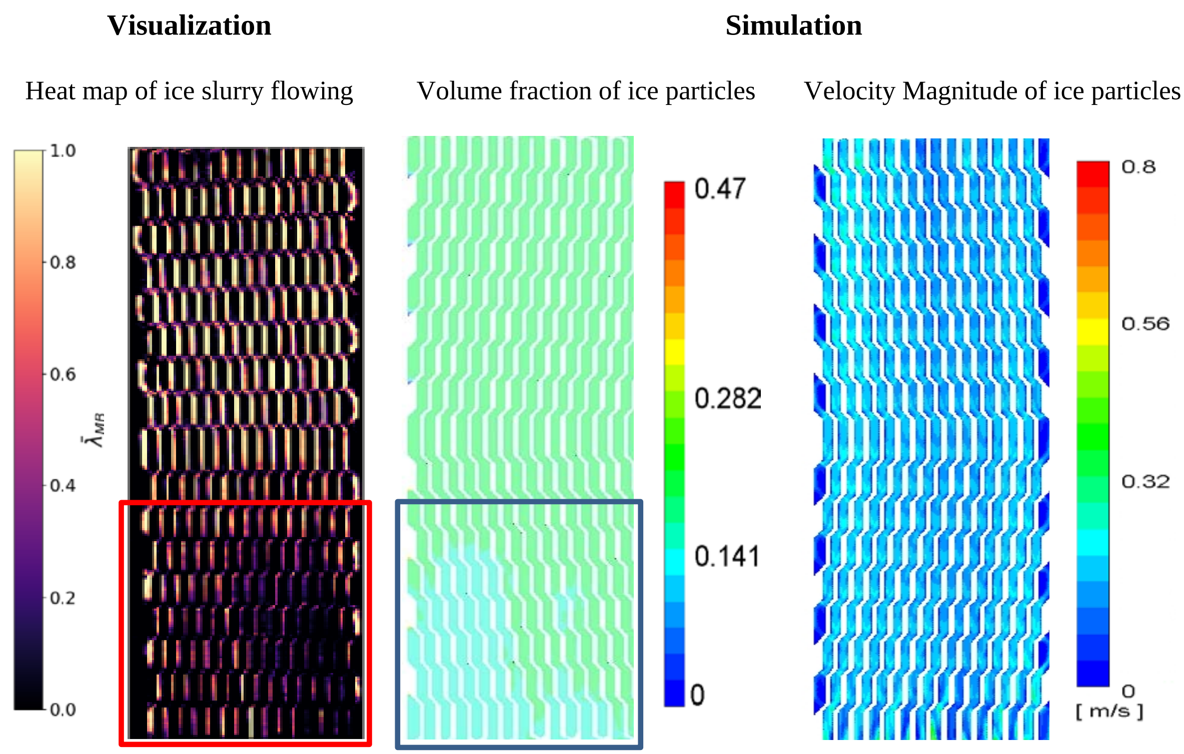

Figure 6: Comparison between the visualization profiles with simulation profiles for ice particles

4. CONCLUSION

The non-isothermal flow of an ice slurry in the special plate heat exchanger from TTZ Company is studied using a 3D Euler-Euler multiphase approach. The approach is based on the kinetic theory of granular flow with the k-ε turbulence model applied to the mixture and takes the effects of particle interaction parameters into account. The predictions from the numerical model are validated with the experimental results from ILK-Dresden for an ice slurry flow in a special plate heat exchanger. The validation of pressure drops shows good agreement between the simulations and experiments results. The predictions of the solid-phase (ice particles) volume fraction and velocity agree with the visualization from the experimental data as well. CFD code is proven to be efficient to predict the behaviour of an ice slurry inside of a plate heat exchanger and it can be used to optimize a new design of special plate heat exchanger for ice slurry flows.

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