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Decarbonation by Energy Saving in Refrigeration and Air-conditioning using Thermal Storage of Green Secondary Refrigerant (Ice Slurries) - Some Studies on Ice Slurries

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Abstract-*In the present research work, a scraped surface ice slurry generator has been designed, developed and fabricated successfully with a focus on collection of experimental data related to ice slurry production and thermal storage, using 10%, 20%, 30% and 40% concentrations of antifreezes Propylene Glycol (PG) and Mono Ethylene Glycol (MEG). R410A (refrigerant blend having zero ODP) is used as primary refrigerant in the primary refrigeration circuit. The lowest slurry temperatures achieved were -20.7 ^{0}C and -25.6 ^{0}C respectively for PG and MEG for 40 % depressant concentration. Thermophysical properties of ice slurry are compared with the chilled water. Three distinct stages- cool down or chilling period, nucleation or unstable ice slurry generation period and stable ice slurry generation period were observed through historical time dependence curves. It was further observed that the freezing temperature reduces with increase in antifreeze mass fraction for PG and MEG.*

Keywords: *depressants, ice slurry, scraped surface ice slurry generator, shell and coil type heat exchanger Performance study.*

1. INTRODUCTION

Refrigeration industry has been continuously working to develop alternative refrigerants having less adverse effects on the environment and having good thermophyscial characteristics, due to some widely used conventional refrigerants which have been identified as harmful for greenhouse substances and also contribute ozone depletion and carbonation. Research work has also been under progress by the thermal storage and use of secondary refrigeration loops to reduce the amount of refrigerant installations. Water or sometimes an aqueous solution is heat transfer fluid in these loops which can be replaced by diphasic secondary refrigerants such as ice slurries to improve system efficiencies, reduce carbonation and also reduce high energy costs, having possibility of enhanced thermal storage and reduction of transport friction losses due to the higher volumetric heat capacity. Ice slurry consists of both a liquid and solid state fraction. The main purpose of using ice slurry is to take advantage of the stored cooling energy (latent heat) in the ice particles during melting. Ice slurry has a great potential for the future due to wide range of applications varying from air-conditioning, commercial refrigeration to industrial production processes, medicine, milk production where high peak loads are to be adjusted. Ice Slurry can be produced and stored in thermal storage systems during off-peak hours and used in peak-hours.

Recent developments in the ice slurry generator technology and advanced design concepts have made this technology a industrially viable alternative to existing conventional secondary refrigeration systems. Presently, the major research work for development of ice slurry generator has taken place

only in developed countries due to very high fixed cost. The scraped surface ice slurry generator is

commercially the most technologically developed and widely accepted ice slurry generation process over the last two decades. From the literature review it is clear that the basic understanding of the governing ice slurry crystallization mechanism is rather limited and is still speculative. Further, at present the design of ice slurry generation system are purely in the hands of ice slurry manufacturer who keep detailed operating data proprietary. Therefore there is an urgent need for better understanding of basic crystallization and heat transfer mechanisms to optimize and develop relatively compact, efficient and less costly ice slurry generators. Ice slurry is a phase-changing secondary fluid consisting of both a liquid state fraction and a solid–state fraction. Depending on the type of additive and additive concentration, the operating temperature for ice slurry can be chosen from 0 to at least -35ºC (Cecilia Hagg et al. 2005; Åke Melinder et al. 2007). Beyond the advantages of the traditional indirect systems for lowering the emissions of refrigerants and refrigeration plants, ice slurry is a more efficient secondary fluid than single-phase fluids. Using ice slurry with accumulation increases the possibility to build indirect systems without increasing the energy consumption. Ice slurry with thermal storage has a great potential for the future due to wide range of industrial applications varying from air-conditioning and commercial refrigeration to industrial production processes medicine and in the milk production where high peak loads are to be adjusted.

In the present research work, a scraped surface ice slurry generator has been designed, developed and fabricated successfully with a focus on collection of experimental data related to ice slurry production and thermal storage using 10%, 20%, 30% and 40% concentrations of antifreezes (PG and MEG). R410A (refrigerant blend having zero ODP) is used as primary refrigerant in the primary refrigeration circuit. Lowest slurry temperatures achieved were -20.7^{0}C, -25.6^{0}C for PG, MEG for 40% depressant concentration. Thermophysical properties of ice slurry compared with chilled water. It was observed that cooling duty found to be higher by using ice slurry instead of chilled water in PHE.

2. DESCRIPTION OF EXPERIMENTAL SETUP

Figure 1 shows a schematic diagram of the experimental facility, consisting mainly of two independent circuits: ice slurry formation circuit and ice slurry flow circuit.

The ice slurry formation circuit consists of a 74 liter capacity scraped surface ice slurry generating system. The major components of this system are: ice slurry generator (1) with spring loaded scraper, condensing unit (9), pumps (2) and a thermal storage tank (4) with an agitator (8). A spring loaded scraper was installed to produce agitation inside the inner tank of ice slurry generator. The functions of scraper are to enhance the heat transfer between the slurry and the refrigerant, accelerate the nucleation, scrape the crystals adhering to the inner tank wall and ensuring the homogeneity of the generated slurry. This system produces fine ice crystals with diameters in the range between 150 and 200 μm.

The ice slurry flow circuit was designed, fabricated and assembled to enable pressure drop and heat transfer data measurements of ice slurry mixtures and water in pipes, bends and heat exchangers. A standard PHE (manufactured by Alfa Laval) (22) normally used in traditional secondary loop systems was tested both thermodynamically and hydraulically with ice slurry and chilled water for wide range of flow. The heat exchanger has 24 plates with 11 channels on each side. Each plate is 480 mm(18.9 inch) in height and 150 mm (5.9 inch) width. The hydraulic diameter is 4 mm (0.16 inch) and the total heat transfer area is 0.6765 m^2 (7.2817 ft^2). The cold fluid was forced to flow through the one set of 11 channels side, while the hot water prepared in a thermally insulated water tank, was used as the cold load and flowed counter-current on the other side. The entire pipe work and the thermal storage tank are well insulated. The various instrumentation used for measurements mass flow meters (14 & 19), thermocouple (20), pressure transducers (21), data acquisition system(10) and sampling points (17) etc. The details of these components are shown in Table 1.

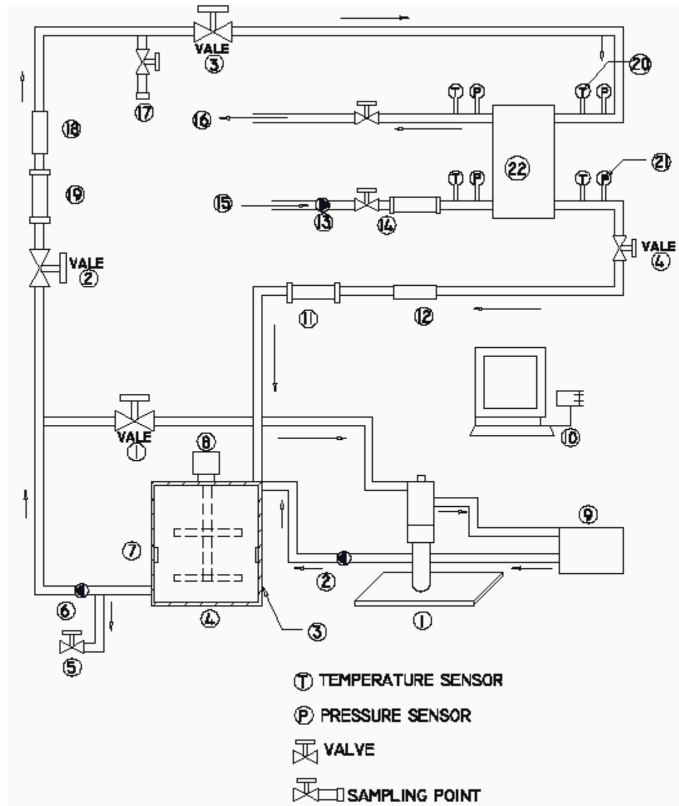


Figure 1 Schematic diagram of the experimental facility (1 = Ice Slurry Generator,

2 = Pump, 3 = Insulation, 4 = Ice Slurry Storage Tank, 5 = Drainage, 6 = Pump,

7 = Thermocouple, 8 = Agitator, 9 = Condensing Unit, 10 = Data Acquisition

System, 11 = Mass Flow Meter, 12 = Transparent Viewing Section, 13 = Pump,

14 = Mass Flow Meter, 15 = Water in, 16 = Water out, 17 = Sampling Point,

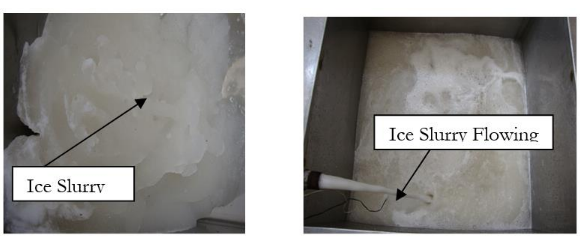
18 = Transparent Viewing Section, 19 = Mass Flow Meter, 20 = Thermocouple,

21 = Pressure Transducer, 22 = Plate Heat Exchanger

Table 1. Specifications of measuring instruments

Instrumentation	Type/make/ model	Range	Accuracy
Mass flow meter	OPTIMASS 8300C S25	0 to 5000 kg/h	± 0.1 % (of reading)
Differential pressure transmitter	SIEMENS	0 to 1000 mbar	± 0.01 bar
Resistance-temperature detectors	LTX-3000/D	-50ºC(-58ºF)to 0ºC(32ºF) to 99 ºC (210.22ºF)	± 0.01 K
Data logger (Data Acquisition System)	DT80-2	16 terminal points	± 0.01 %

The temperatures for both flowing fluids were measured at the inlet and the outlet of the heat exchanger using resistance temperature detectors. One thermocouple was used to monitor the temperature of the mixture in the storage tank. Pressure drop across the heat exchanger streams was measured using pre-calibrated differential pressure transducers which were directly inserted into the fluid. Pressure sensors were located at the inlet and outlet of PHE to record the variation of the inlet and outlet pressures of the tested fluids and the pressure drop was then inferred. Mass flow meters connected at the upstream sides of respective fluids were used to measure the flow rates. All the temperature, pressure and flow rate sensors were connected to a PC based data acquisition system where data were automatically recorded in every 30 seconds for further analysis.



(a) Ice Slurry (b) Ice Slurry flowing in thermal storage tank

Figure 2 Photographs of Ice Slurry

3. EXPERIMENTAL PROCEDURE AND DATA COLLECTION

The plate heat exchanger was tested for water to water flow, and ice slurry to water flow. The hot (primary) fluid in the heat exchanger was water, obtained directly from a storage tank provided with an emersion heater. Chilled water and ice slurry was used as secondary fluid. The secondary fluid flow rate was measured upstream of the heat exchanger using a mass flowmeter while the main hot water flow rate was measured using a mass flowmeter just before the inlet to the heat exchanger. Experimental runs were performed using chilled water (approximately 4 ºC (39.2 ºF)) at flow rates starting from 0.3 m^3/h (300 LPH) to 3.0 m^3/h (3000 LPH). The hot water, at approximately 17.4 °C (63.3 ºF), was then allowed to flow through the heat exchanger and the flow rate was adjusted to obtain a 0.7 m^3/h (700 LPH) of primary fluid flow. The heat transfer results from these runs would form the bases for validating and comparing the ice slurry results. Propylene Glycol (PG) and Mono Ethylene Glycol (MEG) are used as depressants (10%, 20%, 30% and 40% by weight) for formation of ice slurry. Once the percentage of ice in the thermal storage tank reached the desired value, the slurry generation system was shut down and the mixer inside the tank was operated. This allowed proper mixing of the ice/liquid solution, producing a homogeneous mixture throughout the tests. The secondary fluid (ice slurry) was circulated through the heat exchanger by ice slurry circulation pump initially at a flow rate of 0.3 m^3/h (300 LPH) adjusted using valve. Simultaneously the data of temperatures and pressures across the PHE are collected at different flow rates up to the maximum flow rate of 3.0 m^3/h (3000 LPH).

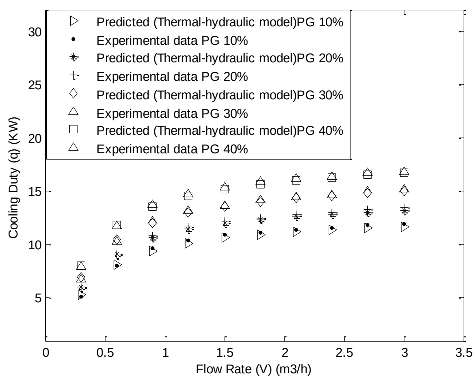
The ice fraction was measured during each test. The flowing mixture was sampled near the inlet of the heat exchanger. The ice crystals were separated from the mixture in separate container and the ice fraction was determined from the corresponding weight of the ice crystals collected separately. The ice fraction was kept 10% by weight for all the runs .In data reduction, calculation of fluid properties was based on the average fluid temperature across each circuit of the heat exchanger. The heat exchanger was insulated using polyurethane insulating foam and heat transfer across the walls to the ambient was neglected. Heat balance between the hot and chilled water sides revealed less than 5% difference between the two values for the range of flows tested.

4. EXPERIMENTAL RESULTS AND DISCUSSION

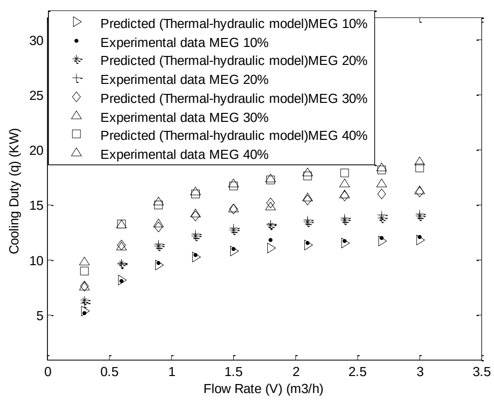
Cooling Duty

Comparison of predicted cooling duty using T-H modeling with experimental data using PG and MEG as antifreezes (10%, 20%, 30% and 40% concentration), and chilled water is shown in Figure 3. It can be seen that predicted ice slurry cooling duty matches reasonably well with the experimental data. For comparison purposes, cooling duty results for water to water are also presented. It can be seen that ice slurry cooling duty is around 25-100% higher than that of chilled water for the selected flow rate. The ice slurry cooling duty increases with flow rate and antifreeze concentrate.

The term "uncertainty analysis" refers to the process of estimating how great an effect the uncertainties in the individual measurements have on the calculated result (Moffat et al.). In the present study, the overall uncertainty is associated with the measurement of the overall heat transfer coefficient and cooling duty for ice slurry and chilled water. The functional dependence of these parameters depends on measured value of inlet and outlet temperatures, mass flow rate and pressure drop. The maximum overall uncertainty in measurement of overall heat transfer coefficient and cooling duty is ±10.3% and ±10.1% respectively. The uncertainty in the measurement of the temperature, flow rate and pressure drop are ±0.35%, ±1.69% and ±0.27% respectively.



(a)



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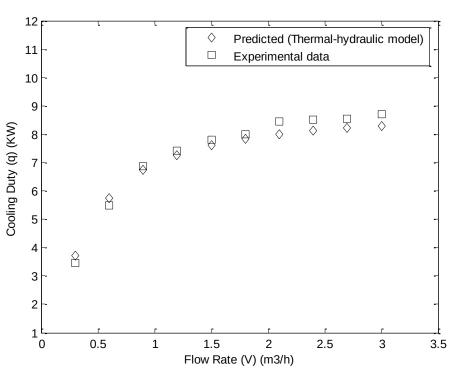


(c)

Figure 3 Flow rate vs cooling duty of (a) Ice Slurry using PG as antifreeze (b) Ice Slurry using MEG as antifreeze (c) Chilled water

Ice Slurry Temperature

The lowest ice slurry temperatures achieved are -5.7 ^{0}C, -10.2 ^{0}C, -15.5 ^{0}C and -20.7 ^{0}C for PG, -6.3 ^{0}C, -12.6 ^{0}C, -19.0 ^{0}C and -25.6 ^{0}C for MEG at 10%, 20%, 30% and 40% concentrations respectively. A summary of these temperatures are given in Table 2.

Table 2. Minimum ice slurry temperature at various concentrations of PG and MEG

Antifreeze	Minimum ice slurry temperature achieved (^{0}C)			
	10%	**20%**	**30%**	**40%**
PG	-5.7	-10.2	-15.5	-20.7
MEG	-6.3	-12.6	-19.0	-25.6

Freezing Temperatures vs. Antifreeze Mass Fraction

Freezing temperatures vs. antifreeze mass fraction is shown in Figure 4. Here, freezing temperature is inversely proportional to antifreeze mass fraction. When water freezes out after the temperature of the liquid mixture has passed below the freezing point, the concentration of the antifreeze increases in the liquid-phase. The increased antifreeze concentration implies that the freezing point of the remaining liquid-phase is further lowered and in order to freeze out more ice the temperature of the mixture has to be further lowered below the current freezing point of the liquid. The result is that the fluid has a freezing range rather than a definitive freezing point. Thus by plotting the freezing point as a function of the antifreeze concentration, one obtains a freezing point curve as a function of the additive mass concentration of different freezing point depressants (Figure 3). The lowering of the temperature of the ice slurry is independent of the effect of the latent heat from the phase change, but dependent on the sensible heat of the mixture. Since it is the advantage of the latent heat in ice slurry that is desired, one desires a liquid mixture where the latent heat dominates. To minimize the influence of the sensible heat, a fluid with a relatively low first derivative of the freezing point curve (flat freezing point curve) is to be preferred.

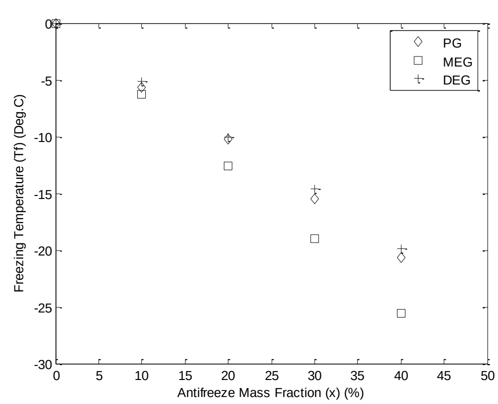
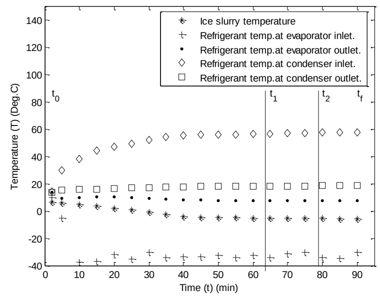
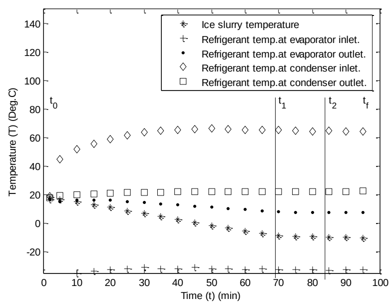


Figure 4 Freezing curve of water-PG and water- MEG mixture

Recorded temperatures of aqueous solution of antifreezes, refrigerant temperatures at evaporator inlet and outlet, refrigerant temperatures at condenser inlet and outlet at different concentrations are plotted for better understanding of the system (Figures. 5(a) to 5(d) for PG and Figures 6(a) to 6(d) for MEG) with respect to freezing time required for ice slurry formation.



(a)



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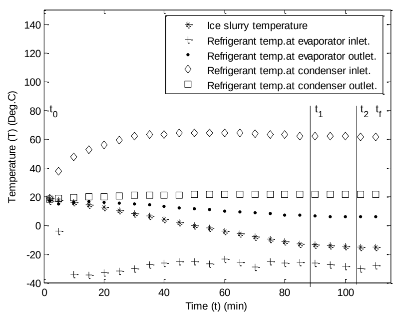

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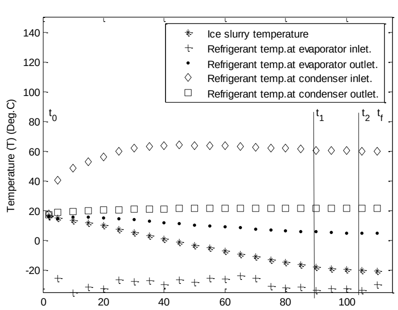
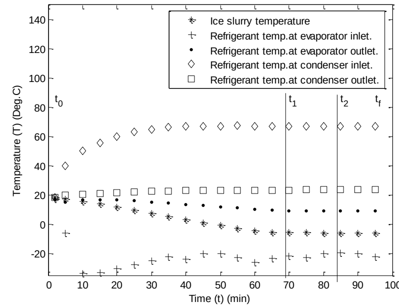
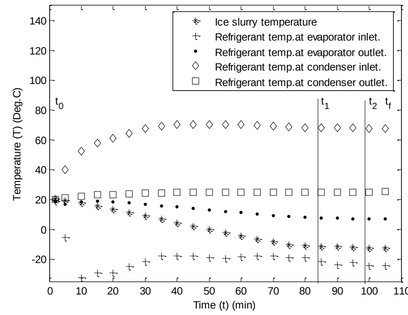


(d)

Figure 5 Freezing temperature vs time for PG at (a)10 %, (b)20 %, (c)30 % and (d)40 % concentration



(a)



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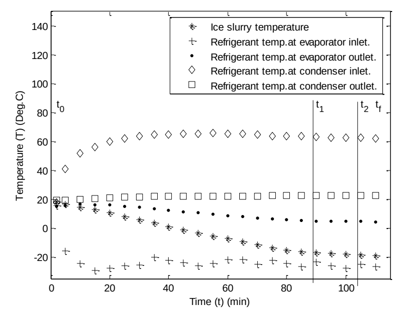

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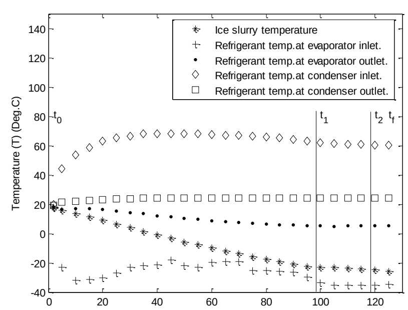

(d)

Figure 6 Freezing temperature vs time for MEG at (a)10 %, (b)20 %, (c)30 % and (d)40 % concentration

From the present experimental ice slurry generation data it can be observed that ice slurry generation process can be divided into three stages- cool down or chilling period, nucleation or unstable ice slurry generation period and stable ice slurry generation period. The first stage (cool down period) starts t_0 to t_1, where t_0 is the starting time of the experiment and t_1 is the time at the end of the chilling period which is the on-set of the super-cooling phenomenon. During the chilling period volumetric ice concentration is zero. As observed in Figures 5(a) to 5(d), the freezing temperature reduces with increase in antifreeze mass fraction for PG and MEG solution initially chilled continuously without phase change in stage 1. First phase time duration is 64, 69, 87 and 90 minutes respectively for 10%, 20%, 30%and 40% concentration of PG. Similar trends were observed for MEG (Figures 6(a) to 6(d)) but first phase time duration was relatively higher as compared to PG. During this stage the average evaporator temperature decreases sharply which causes increase in the refrigeration capacity and compressor work. Therefore, the condenser inlet temperature increases due to higher heat rejection quantity. The second stage (nucleation period) starts from t_1 to t_2, where the ice seeds after the super cooling phenomenon is observed and the volumetric ice concentration increases till its maximum value at the end of this period (at t_2). In stage 2, nucleation of ice particles occurs and it is characterized by 0.5 to 1^0C sudden increase in temperature of the process fluid due to the release of the fusion heat of ice. Finally the third stage (ice slurry generation period) starts from t_2 to the end of the experiment, at t_f. During this stage the ice concentration is maintained constant at its maximum value. During stage 3 the heat transfer is affected by the release of the latent heat of water freezing.

The term “uncertainty analysis” refers to the process of estimating how great an effect the uncertainties in the individual measurements have on the calculated result (Moffat et al.). In the present study, the uncertainty in the measurement of the freezing temperature, is $\pm$0.30%.

5. CONCLUSIONS

1. The minimum ice slurry temperatures achieved are -5.7 ^{0}C, -10.2 ^{0}C, -15.5 and -20.7 ^{0}C for PG and -6.3 ^{0}C, -12.6 ^{0}C, -19.0 ^{0}C and -25.6 ^{0}C for MEG at 10%, 20%, 30% and 40% antifreeze concentrations respectively.
2. By flowing ice slurry from thermal storage tank, in place of chilled water in plate heat exchanger, cooling duty found to be higher and pressure drop is slightly higher in case of ice slurry. Cooling duty of ice slurry increases with increase in antifreeze concentrations 10%, 20%, 30% and 40%. Cooling duty found to be increased by 50% at 0.3m^3/h(300 LPH) and increased by 37 % at 3.0 m^3/h(3000 LPH) of 10% ice crystal PG ice slurry and pressure drop increased by 10% at 0.3m^3/h(300 LPH) and increased by 7 % at 3.0 m^3/h(3000 LPH).
3. For water to ice slurry from thermal storage, experiments in PHE, cooling duty for ice slurry stream increase with increase in flow rate of ice slurry.
4. It is observed that the freezing temperature reduces with increase in antifreeze mass fraction for PG and MEG.
5. Three distinct stages- cool down or chilling period, nucleation or unstable ice slurry generation period and stable ice slurry generation period were observed through historical time dependence curves.

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